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PROCESS INTEGRATION OF ABSORPTION HEAT PUMPS

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PROCESS INTEGRATION OF ABSORPTION HEAT PUMPS

présentée par: BAKHTIARI Bahador

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a été dûment acceptée par le jury d'examen constitué de :

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DEDICATION

To my wife, my beloved parents and my sister

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RÉSUMÉ

Dans un contexte global, il est essentiel de réduire la consommation de combustibles fossiles afin de diminuer l'impact des activités humaines sur l'environnement. Le milieu industriel consommant beaucoup d'énergie, la réduction de son utilisation dans ce secteur est indispensable. Différentes options sont possibles pour réduire l'utilisation de l'énergie en milieu industriel. Les mesures proposées incluent l'intégration de pompes à chaleur, la cogénération de puissance et de chaleur, l'utilisation de différents combustibles, des opérations unitaires plus efficaces et l'augmentation de la récupération de l'énergie par échange de chaleur. Les pompes à chaleur (PC) sont des technologies de conversion de l'énergie qui sont utilisées pour augmenter la qualité de l'énergie en augmentant la température à laquelle celle-ci est disponible. Les pompes à chaleur à absorption (PACA) émergent comme une alternative potentielle aux pompes à chaleur à recompression de vapeur (PCRV) qui sont plus courantes. Elles sont mues thermiquement et lorsque qu'elles sont judicieusement positionnées dans un procédé industriel, elles peuvent être exploitées sans pratiquement acheter d'énergie.

L'analyse de pincement est une technique utilisée pour maximiser la récupération de la chaleur interne dans un procédé. La thermodynamique de l'analyse de pincement dicte une règle fondamentale: il ne doit y avoir aucun transfert de chaleur de part et d'autre du point de pincement. Si cela se produit, le procédé subira une double pénalité, soit l'accroissement simultané des demandes en refroidissement et en chauffage. D'autre part, une pompe à chaleur doit transférer la chaleur d'un côté à l'autre du point de pincement : elle doit être intégrée de manière à ce que la source de chaleur soit située où se trouve un excès d'énergie, c'est-à-dire sous le pincement, et le puits de chaleur soit situé où il y a un besoin en énergie, c'est-à-dire au-dessus

du pincement. La méthodologie d'intégration de certains types de PC, tels la PCRV ou les pompes à chaleur à compression alimentées électriquement est bien définie dans la littérature. Par contre, l'intégration d'une configuration plus complexe combinant des PACAs et des transformateurs de chaleur à absorption n'a pas été étudiée en profondeur.

Par conséquent, le premier objectif de ces travaux de recherche est l'élaboration d'une nouvelle méthodologie pour le positionnement et l'évaluation appropriés des pompes à chaleur à absorption. Des lignes directrices sont formulées pour la sélection des sources et des puits de chaleur qui optimisent l'intégration des PACAs tout en respectant les contraintes du procédé et les conditions de fonctionnement. La méthode prend pour hypothèse qu'une analyse de pincement a été réalisée et que les données thermiques de tous les courants sont disponibles. Puisque les pompes à chaleur sont plus coûteuses que les échangeurs de chaleur, l'échange de chaleur doit être considéré en premier lieu et le procédé doit être intégré dans cette optique. Il convient de signaler que d'autres options de récupération d'énergie tels que la récupération des condensats et la fermeture des circuits d'eau sont recommandés avant de réaliser l'analyse de pincement et le placement de la pompe à chaleur. En général, ces options sont des mesures de récupération d'énergie plus économiques et elles contribuent à diminuer la température de pincement, ce qui facilite l'implantation de pompes à chaleur à absorption.

La méthode développée propose certaines améliorations qui permettent d'obtenir des configurations supérieures à celles qui peuvent être générées en utilisant les deux méthodes traditionnelles pour la conception des PACAs et l'intégration des pompes à compression. En effet, cette méthode considère l'interaction entre la PACA et le procédé. Elle permet de déterminer les flux qui correspondent le mieux à la PACA, la meilleure configuration et le couple de travail approprié, en considérant les diagrammes de phases de fluides ainsi que d'autres contraintes techniques et de conception. Les avantages et les résultats de la méthodologie sont illustrés en utilisant un exemple d'implémentation d'une PACA dans un procédé de mise en pâte kraft. Il a été conclu que 8 configurations différentes sont possibles d'un point de vue thermodynamique et pratique. Deux cas pratiques, soit la PACA d'un seul circuit de type II et soit la PACA à double flux de type II ont été choisis pour une conception et une évaluation économique détaillés. Les coûts d'installation calculés sont de 3.4 M$ pour la PACA à simple effet et 2.6 M$ pour la PACA à double effet de type II, ce qui représente un temps de recouvrement de 1 et 1.2 ans respectivement. Pour le cas considéré, la PACA d'un seul circuit de type II est la meilleure configuration des deux cas, car elle a la même période de recouvrement, une meilleure performance à long terme tout en étant de conception plus simple.

Le deuxième objectif de ces travaux de recherche, était d'identifier des opportunités pour une intégration optimale des PACAs dans l'industrie des pâtes et papiers (P&P). L'industrie des P&P est parmi les industries principales du Canada, elle est caractérisée par des demandes énergétiques, sous forme d'électricité et d'énergie thermique, très importantes (27% de la consommation industrielle totale). Deux opportunités d'intégration des PACAs ont été identifiées dans le cadre de ces travaux de recherche. Il a été démontré que même dans le cas d'une usine où la consommation d'eau et d'énergie sont optimisées, il est possible d'économiser davantage d'utilitaires. La première opportunité est liée à la production d'eau froide à 6 °C dans l'atelier de fabrication de dioxyde de chlore (ClO_2). Grâce à la méthodologie développée au cours de ces travaux de recherche, il a été démontré qu'un refroidisseur à double effet peut être implanté dans l'atelier de fabrication de produits chimiques pour produire simultanément de l'eau froide et de l'eau chaude en utilisant de la vapeur à moyenne pression (MP) comme énergie motrice. La réduction nette de la consommation de vapeur est de 3.6 MW, dans la configuration actuelle du procédé, 2 MW de vapeur à MP sont utilisés pour produire de l'eau chaude et 1.6 MW - pour générer de l'eau froide. En même temps, le besoin de 139 kg/s d'eau de refroidissement et 0.26 MW d'électricité a été éliminé. La deuxième opportunité a aussi été identifiée dans l'atelier de blanchiment de la pâte. Dans la configuration actuelle du procédé, les effluents du blanchiment sont envoyés aux égouts. Une source de chaleur au-dessous du point de pincement peut être utilisée pour une PACA. Il a été démontré qu'une combinaison de puits de chaleur au-dessus du point de pincement doit être considérée. Une pompe à chaleur à un seul effet a été sélectionnée pour concentrer la liqueur noire et produire de l'eau chaude en valorisant la chaleur des effluents du blanchiment et en utilisant de la vapeur à MP comme énergie motrice. La réduction nette de consommation de vapeur est de 1.92 MW. La réduction nette des besoins en énergie de chauffage et de refroidissement est de 2 % et 8.6 % respectivement.

Dans la troisième partie du projet, une analyse expérimentale et par simulation d'une PACA à un seul effet H_2O-LiBr dont la capacité de refroidissement est de 14 kW, a été réalisée. Un modèle de conception et de dimensionnement de PACAs à H_2O-LiBr a été développé et les données expérimentales de sa performance ont été comparées avec les résultats de la simulation. Le modèle a deux champs d'application. Il peut être utilisé soit pour analyser la performance d'une installation soumise à différentes conditions opératoires ou soit pour faire le design et le dimensionnement d'une nouvelle installation.

Un plan expérimental a été conçu pour mesurer la performance de l'appareil, caractérisé par son COP et sa capacité de refroidissement, sous différentes conditions d'opération et en utilisant comme facteurs dominants les températures d'eau chaude, d'eau réfrigérée et d'eau de refroidissement et les débits d'eau chaude et d'eau de refroidissement. Pour chaque groupe d'expériences, les données expérimentales et simulées sont présentées et il y a une bonne concordance entre les deux types de valeurs. Il a été montré que la pompe à chaleur maintenait un grand COP pour une plage étendue de températures d'entraînement, ce qui indique une utilisation efficace de la source de chaleur et en fait une bonne candidate pour la combiner avec des collecteurs solaires qui fourniraient l'énergie motrice au générateur et des unités de cogénération. Le COP varie très peu pour toute l'étendue des températures de l'eau réfrigérée. La capacité de refroidissement et le COP sont améliorés par l'utilisation de hauts débits d'eau de refroidissement.

Dans la dernière partie de ce projet, il a été montré comment un modèle développé peut être combiné avec la méthodologie de mise en œuvre du dimensionnement d'une PAC sur un procédé existant. Le modèle est utilisé pour calculer les paramètres de conception et d'opération d'une PAC intégré à un procédé Kraft. Dans l'exemple sélectionné, l'objectif de la pompe à chaleur est de valoriser la charge thermique basse température de l'effluent de la section de blanchiment et d'augmenter sa température. De l'eau chaude a été fournie à une température supérieure à la température de pincement. De la vapeur basse pression à 345 kPa a été utilisée comme énergie d'entrainement pour le générateur. A partir du modèle, les paramètres de conception de la pompe à chaleur ont été calculés en régime permanent.

ABSTRACT

In a global context, it is essential to reduce the consumption of fossil-based energy in order to decrease the environmental impact of human activities. Industry's share of the total energy consumption is very large, and thus reduction of energy use is highly motivated in this sector. A number of different options are possible to reduce industrial energy use. Proposed measures include e.g. heat pumping, combined heat and power generation, fuel switching, more efficient unit operations, and increased heat recovery by heat exchange. Heat pumps are energy conversion devices that are used to upgrade the quality of heat by raising the temperature at which it is available. Absorption heat pumps (AHP) are emerging as a potential alternative to the more common vapour recompression heat pumps (VRHP). They are thermally driven and when judiciously positioned into an industrial process, they can be operated with practically no purchased power.

Pinch Analysis is a technique used to maximize internal heat recovery within a process. The thermodynamics of Pinch Analysis dictates a fundamental rule: there must not be transfer of heat from above to below the pinch point. If this happens, the process suffers a double penalty: the simultaneous increase of the cooling and heating requirements of the process. On the other hand, an HP must transfer heat in the opposite direction from below to above the pinch point; so it should be integrated in such a way that the heat source is situated where there is an excess of heat, i.e. below the pinch, and the heat sink where there is a need for heat, i.e. above the pinch The methodology of integration of traditional heat pumps such as vapour recompression heat pumps or electrically driven compression heat pumps is well known and discussed in the

literatures. However integration of more complex configurations like absorption heat pumps and absorption heat transformers in a process has not been thoroughly investigated.

Thus, as the first aim of this research, a new methodology for the appropriate placement and evaluation of absorption heat pumps in the process is presented. Guidelines are formulated for the selection of heat sources and sinks that maximize the benefit derived from the heat pumping while respecting the process constraints and the operating requirements of the absorption heat pump. The method assumes that a Pinch Analysis of the process has been performed and thermal data on all streams are available. Since heat pumping is more expensive than heat exchanging, the HEN should first be considered and the process should be heat-integrated. It should be mentioned that other energy recovery methods such as condensate return and water closure are recommended before pinch analysis and heat pump implementation. In general, they are cheaper energy recovery measures and they help reduce the pinch point temperature, which facilitates the implementation of AHPs.

The proposed method introduces several features that reveal superior designs to what could be achieved by either of the two conventional methods for designing AHPs and integration of traditional compression-based heat pumps. Indeed, it considers the interactions between the AHP and the process. It determines the streams that will best fit the AHP, and the best configuration and working pair, taking into account working fluid phase diagrams and other design and technical constrains. The advantages and outputs of the methodology are exemplified using an AHP implementation in a Kraft pulping process. It is found that there are 8 different thermodynamically and practically feasible configurations. The two most realistic cases; which are a single stage AHP type II and a double lift AHP type II have been chosen for detailed design and priliminary economic evaluation. The installed costs are calculated as 3.4 and 2.6 M$ for the single stage and double lift AHP type II, which give the simple payback times of 1 and 1.2 years, respectively. The single stage AHP type II is found to be the best configuration for the actual case considered because of almost identical SPB, better performance in the long term and much simpler design.

As the second aim of this research, the opportunities for the optimal integration of AHPs in the pulp and paper (P&P) industry are identified. The pulp and paper industry (P&P) is Canada's premier industry characterized by very large energy requirements in the form of electricity and thermal energy (27% of all industrial energy consumption). In this research, two generic opportunities were identified. It is shown that even for a fully energy and water optimized mill, there is still a potential for further utility savings. The first identified opportunity deals with the production of chilled water at 6 °C in the chlorine dioxide (ClO_2) making plant. Considering the methodology developed in this research, it is shown that a double lift chiller can be implemented in the bleaching chemical making plant to produce chilled and hot water simultaneously, using MP steam as the driving energy. The net reduction of steam consumption is 3.6 MW. Also, the need for 139 kg/s of cooling water and 0.26 MW of electricity is eliminated. The second opportunity is also identified in the pulp bleaching plant. Under current process configuration, bleaching effluents is sent to the sewer. It is a heat source below the pinch point that could be used for an AHP. It is shown that a combination of heat sink streams above the pinch point should be selected. A single stage heat pump is selected to concentrate the black liquor and produce useful hot water by upgrading heat from the bleaching effluent and using MP steam as driving energy. The net reduction of steam consumption is 1.92 MW. The net hot and cold energy demand reductions represent 2 and 8.6 % of the MHR and MCR, respectively.

As the third part of the project, an experimental and simulation analysis of a laboratory single-stage H_2O-LiBr absorption heat pump with a cooling capacity of 14 kW has been performed. A design and dimensioning model of H_2O-LiBr absorption heat pumps was developed and experimental measurements of its performance were compared with simulation results. The model has two different fields of applications. It can be used either to analyze the performance of an existing machine under various operating conditions or to design and dimension a new one.

An experimental plan is designed, aimed at measuring the performance of the machine, as described by the COP and the cooling capacity, under different operating conditions using the temperature of chilled, cooling and hot water and the flow rate of cooling water and hot water as the dominant factors. For each set of experiments, experimental and simulation data are presented and there are close agreement between them. It is shown that the heat pump has a high COP over a large driving temperature range, which indicates an efficient use of the heat source and makes it a good candidate for linkage with simple flat plate solar collectors and cogeneration units. The COP varies very little over the experimental range of chilled water temperature. The cooling capacity and the COP are improved by high cooling water flow rates.

In the last part of this research, it is shown how the developed model can be linked with the AHP implementation methodology for the dimensioning of an AHP implemented in an exciting process. The model is used to calculate the design and operating parameters of an AHP implemented in a Kraft process. In the selected example, the particular objective of the heat pump is to upgrade the low temperature heat load from the effluent of the pulp bleaching plant to a higher temperature level. Hot water from the power plant was selected as the proper above the pinch point. Low pressure steam (LP) at 345 kPa was used as the driving energy for the generator. The model calculated the design parameters of the heat pump in a steady state operation.

CONDENSÉ EN FRANÇAIS

Dans un contexte global, il est essentiel de réduire la consommation de combustibles fossiles afin de diminuer l'impact des activités humaines sur l'environnement. Le milieu industriel consommant beaucoup d'énergie, la réduction de son utilisation dans ce secteur est indispensable. Différentes options sont possibles pour réduire l'utilisation de l'énergie en milieu industriel. Les mesures proposées incluent l'intégration de pompes à chaleur (PC), la cogénération de puissance et de chaleur, l'utilisation de différents combustibles, des opérations unitaires plus efficaces et l'augmentation de la récupération de l'énergie par échange de chaleur.

Les pompes à chaleur sont des technologies de conversion de l'énergie qui sont utilisées pour augmenter la qualité de l'énergie en augmentant la température à laquelle celle-ci est disponible. Les pompes à chaleur sont installées avec succès dans plusieurs secteurs domestiques et commerciaux pour satisfaire simultanément des demandes en refroidissement et en chauffage. Dans le milieu industriel, l'application de pompe à chaleur est plus rare. Les pompes à chaleur industrielles offrent de nombreuses opportunités d'application pour certains procédés. Plusieurs procédés industriels sont caractérisés par des temps de fonctionnement très longs qui augmentent le potentiel de conservation de l'énergie pour les pompes à chaleur. On dit souvent que la consommation énergétique des industries pourrait éventuellement diminuer de 20% si des mesures d'efficacité énergétique étaient appliquées. Augmenter l'efficacité énergétique d'un procédé n'est pas le seul avantage des pompes à chaleur, puisque celles-ci permettent également de résoudre des problèmes de production et des problèmes environnementaux. Les pompes à chaleur sont des technologies intéressantes pour réduire de façon efficace et économique les émissions de gaz de combustion.

Les pompes à chaleur à absorption (PACA) représentent une alternative potentielle aux pompes à chaleur à recompression de vapeur (PCRV) qui sont plus courantes. Les PACAs utilisent l'effet de la pression sur un cycle d'absorption-désorption d'une solution pour créer une augmentation de la température. Elles sont mues thermiquement et lorsque qu'elles sont judicieusement positionnées dans un procédé industriel, elles peuvent être exploitées en utilisant peut d'énergie achetée. Elles utilisent des fluides thermodynamiques qui ont peu d'impact sur l'environnement. Cependant, elles sont plus complexes que les PCRVs et requièrent un investissement initial substantiellement plus élevé. Dans un contexte d'augmentation des coûts de l'énergie primaire et de législations relatives aux émissions de gaz à effet de serre (GES), elles peuvent représenter une approche économiquement attrayante à l'égard du développement durable.

L'analyse de pincement est une technique utilisée pour maximiser la récupération de la chaleur interne dans un procédé. La thermodynamique de l'analyse de pincement dicte une règle fondamentale: il ne doit y avoir aucun transfert de chaleur de part et d'autre du point de pincement. Si cela se produit, le procédé subira une double pénalité, soit l'accroissement simultané des demandes en refroidissement et en chauffage. D'autre part, une pompe à chaleur doit transférer la chaleur d'un côté à l'autre du point de pincement : elle doit être intégrée de manière à ce que la source de chaleur soit située où se trouve un excès d'énergie, c'est-à-dire sous le pincement, et le puits de chaleur soit situé où il y a un besoin en énergie, c'est-à-dire au-dessus du pincement. La méthodologie d'intégration de certains types de PC, tels la PCRV ou les pompes à chaleur à compression alimentées électriquement est bien définie dans la littérature. Par contre, l'intégration d'une configuration plus complexe combinant des PACAs et des transformateurs de chaleur à absorption n'a pas été étudiée en profondeur.

Par conséquent, le premier objectif de ces travaux de recherche est l'élaboration d'une nouvelle méthodologie pour le positionnement et l'évaluation appropriés des pompes à chaleur à absorption. Des lignes directrices sont formulées pour la sélection des sources et des puits de chaleur qui optimisent l'intégration des PACAs tout en respectant les contraintes du procédé et les conditions de fonctionnement. La méthode prend pour hypothèse qu'une analyse de pincement a été réalisée et que les données thermiques de tous les courants sont disponibles. Puisque les pompes à chaleur sont plus coûteuses que les échangeurs de chaleur, la récupération de chaleur doit être considérée en premier lieu et le procédé doit être intégré en ce sens. Il convient de signaler que d'autre options d'économie d'énergie tels que la récupération des condensats et la fermeture des circuits d'eau sont recommandés avant de réaliser l'analyse de pincement et le placement de la pompe à chaleur. En général, ces options sont des mesures de récupération d'énergie plus économiques et elles contribuent à diminuer la température de pincement, ce qui facilite l'implantation de pompes à chaleur à absorption.

Le condenseur et l'absorbeur doivent libérer leur chaleur au dessus du point de pincement pour réduire l'exigence minimale de chauffage (EMC); le générateur qui est à une température plus élevée ne peut alors qu'être au-dessus du point de pincement et, pour réduire l'exigence minimale de refroidissement (EMR), l'évaporateur doit être en-dessous. Le gain global d'énergie est effectivement le même que pour une PCRV d'une puissance égale, mais aucun travail mécanique est nécessaire.

L'implantation de pompes à chaleur en mode rétrofit peut être décrite par les étapes suivantes:

1. Sélection des flux de procédé potentiellement utilisables par une pompe à chaleur

2. Production d'une combinaison de différents flux (flux froid au dessus du point de pincement et flux chaud en dessous du point de pincement)
3. Estimation des températures des composants de la pompe à chaleur
4. Calcul des charges thermiques de la pompe à chaleur
5. Identification des flux appropriés pour le générateur (type I) et le condenseur (type II)
6. Identification de scénarios thermodynamiquement réalisables
7. Dimensionnement et calcul des coûts
8. Reconfiguration du réseau d'échangeurs de chaleur (si nécessaire)

La méthode développée propose certaines améliorations qui permettent d'obtenir des configurations supérieures à celles qui peuvent être générées en utilisant les deux méthodes traditionnelles pour la conception des PACAs et l'intégration des pompes à compression. En effet, cette méthode considère l'interaction entre la PACA et le procédé. Elle permet de déterminer les flux qui correspondent le mieux à la PACA, la meilleure configuration et le couple de travail approprié, en considérant les diagrammes de phases des fluides ainsi que d'autres contraintes techniques et de conception. Les avantages et les résultats de la méthodologie sont illustrés en utilisant un exemple d'implantation d'une PACA dans un procédé de mise en pâte kraft. L'usine étudiée produit 700 t/d de pâte kraft blanchie de haute qualité. Le besoin moyen de puissance est de 161 MW (19.1 GJ/ADT) produits sous forme de vapeur à haute pression (HP) par quatre chaudières : deux chaudières de récupération de liqueur, une chaudière à biomasse et une chaudière à mazout. Une température d'approche minimum (ΔT_{min}) de 10 ˚C a été choisie pour 61 flux sélectionnés comme des sources ou des puits de chaleur. La température de pincement a été évaluée à 71 ˚C. Les températures chaude et froide de pincement sont de 76˚C et 66 ˚C, respectivement. Les exigences réelles en chauffage et en refroidissement du procédé sont égales à 178 MW et 62.5 MW alors que les exigences minimum de chauffage et de refroidissement (EMC) et (EMR) sont de 122.8 MW et 10 MW, respectivement. Le courant de vapeur condensée dans l'évaporateur 2 dont la température (75.9 °C) est au-dessous du point de pincement et qui porte une charge de chaleur (12.2 MW) a été sélectionné comme source potentielle de chaleur pour la PACA. En considérant les données du procédé et la méthodologie, la liqueur dans les évaporateurs 2 et 3 et la production d'eau chaude dans le déaérateur, ont été considérés comme 3 puits potentiels de chaleur pour la PACA. Dans cet exemple, 6 configurations différentes (un seul circuit, effet double et double flux, PACA du type I ou II) pour les deux couples de fluide de travail (LiBr-H_2O et NH_3-H_2O) ont été analysées, ce qui représente 36 combinaisons. En considérant les contraintes techniques et de conception, certains des cas ne sont pas viables et ont été éliminés. Il a été conclu que 8 configurations différentes sont possibles d'un point de vue thermodynamique et pratique. Deux cas pratiques, soit la PACA à simple effet de type II et soit la PACA à double effet de type II ont été choisis pour une conception et une évaluation économique détaillés. À cette fin, un modèle de simulation simple d'une PACA à simple effet et d'une PACA à double effet de type II utilisant le couple de travail LiBr/H_2O a été développé. Les coûts d'installation calculés sont de 3.4 M$ pour la PACA d'un seul circuit et 2.6 M$ pour la PACA à double flux de type II, ce qui représente un temps de recouvrement de 1 et 1.2 ans respectivement. La valeur actualisée nette a été déterminée afin de démontrer que les deux options sont intéressantes à court terme mais aussi à plus long terme. Pour le cas considéré, la PACA d'un seul circuit de type II est la meilleure configuration des deux cas, car elle a la même période de recouvrement, une meilleure performance à long terme tout en étant de conception plus simple.

Le deuxième objectif de ces travaux de recherche, était d'identifier des opportunités pour une intégration optimale des PACAs dans l'industrie des pâtes et papiers (P&P). L'industrie des P&P est parmi les industries principales du Canada, elle est caractérisée par des demandes énergétiques, sous forme d'électricité et d'énergie thermique, très importantes (27% de la consommation industrielle totale). Une usine de mise en pâte Kraft de construction ancienne consomme en moyenne 25 GJ/t (tonne de pâte séchée au four), tandis qu'une usine moderne consomme 12 GJ/t d'énergie thermique. Les coûts énergétiques représentent un tiers des coûts de production dans les usines canadiennes. L'industrie des P&P connait une compétition intense depuis un certain temps, par conséquent la consommation d'énergie élevée représente désormais un problème à résoudre. De plus, le Canada est un des pays ayant signé le protocole de Kyoto, dont l'objectif est de réduire les émissions de gaz à effet de serre (GES) de 6% par rapport aux valeurs de 1990. Pour cette raison, l'industrie des P&P doit multiplier les efforts, améliorer son efficacité énergétique et diminuer par conséquent la consommation d'énergie thermique et les émissions de gaz. L'implantation des pompes à chaleur à absorption est un moyen adéquat pour atteindre ces objectifs.

Deux opportunités d'intégration des PACAs ont été identifiées dans le cadre de ces travaux de recherche. Il a été démontré que même dans le cas d'une usine où la consommation d'eau et d'énergie sont optimisées, il est possible d'économiser davantage d'utilitaires. La première opportunité est liée à la production d'eau froide à 6 °C dans l'atelier de fabrication de dioxyde de chlore (ClO_2). Dans la configuration actuelle de l'usine, l'eau froide est générée par une pompe à chaleur à compression qui consomme 260 kW d'électricité et 0.6 kg/sec de vapeur à moyenne pression (MP). Grâce à la méthodologie développée au cours de ces travaux de recherche, il a été démontré qu'un refroidisseur à double effet peut être implanté dans l'atelier de fabrication de produits chimiques pour produire simultanément de l'eau froide et de l'eau chaude en utilisant de la vapeur à MP comme énergie motrice. La réduction nette de la consommation de vapeur est de 3.6 MW, dans la configuration actuelle du procédé, 2 MW de vapeur à MP sont utilisés pour produire de l'eau chaude et 1.6 MW - pour générer de l'eau froide. En même temps, le besoin de 139 kg/s d'eau de refroidissement et 0.26 MW d'électricité a été éliminé. L'implantation de la PACA réduit les émissions de GES de 5300 t/a de CO_2, 130 kg/a de CH_4 et 455 kg/a de NO_x.

La deuxième opportunité a aussi été identifiée dans la section du blanchiment. Dans la configuration actuelle du procédé, les effluents du blanchiment sont envoyés aux égouts. Une source de chaleur au-dessous du point de pincement peut être utilisée pour une PACA. Il a été démontré qu'une combinaison de puits de chaleur au-dessus du point de pincement doit être considérée. Une pompe à chaleur d'un seul circuit à été sélectionnée pour concentrer la liqueur noire et produire de l'eau chaude en valorisant la chaleur des effluents du blanchiment et en utilisant de la vapeur à MP comme énergie motrice. La réduction nette de consommation de vapeur est de 1.92 MW, soit de la vapeur à MP qui dans la configuration actuelle du procédé sert à concentrer la liqueur noire et produire de l'eau chaude. La réduction nette des besoins en énergie de chauffage et de refroidissement est de 2% et 8.6% respectivement. L'implantation de la PACA proposée réduit les émissions de GES de 2560 t/a de CO_2, 65 kg/a de CH_4 et 228 kg/a de NO_x.

Dans la troisième partie du projet, une analyse expérimentale et par simulation d'une PACA à simple effet H_2O-LiBr d'une capacité de refroidissement de 14 kW, a été réalisée. Un modèle de conception et de dimensionnement de PACAs à H_2O-LiBr a été développé et les données expérimentales de sa performance ont été comparées avec les résultats de la simulation. Le système à absorption comprend quatre échangeurs de chaleur principaux : condenseur, générateur, absorbeur et évaporateur, ainsi qu'un cinquième échangeur auxiliaire placé entre le générateur et l'absorbeur, l'échangeur de chaleur de la solution (ECS). Les quatre échangeurs principaux sont de type à tubes et calandre, comprenant des tubes horizontaux et un arrangement où les pertes de pression sont négligées afin de permettre la distribution libre de la solution de travail et du réfrigérant. L'échangeur de chaleur de la solution est un échangeur à plaques, disponible commercialement. Le système a été conçu à des fins de refroidissement, soit pour produire de l'eau froide pour les systèmes de climatisation qui nécessitent une température à l'évaporateur entre 6 et 12 °C, soit pour fournir un refroidissement aux plafonds chauffants avec une température à l'évaporateur entre 15 et 18 °C. L'absorbeur et le condenseur sont refroidis à l'eau. Pour les conditions nominales, l'eau de refroidissement est chauffée de 27 à 40.4 °C. La chaleur motrice est fournie à 95 °C, ce qui rend le système approprié pour être utilisé dans des unités de cogénération, entrainées par des moteurs ou des piles à combustible, ou dans des systèmes de chauffage par énergie solaire ou d'air conditionné. La centrale électrique est implantée pour assurer des conditions constantes de températures et de débits, nécessaires dans les circuits externes. Elle est reliée à la pompe à chaleur par trois boucles d'eau. Le prototype est contrôlé par ordinateur à l'aide du programme Labview. Il est équipé de 23 capteurs pour enregistrer les pressions, les températures et les débits du procédé.

Un modèle analytique pour simuler une pompe à chaleur à absorption mono-étagée de type I H_2O-LiBr a été développé, l'objectif étant d'apporter une aide à la conception et au dimensionnement de ce type d'appareils thermiques. Le modèle a deux champs d'application. Il peut être utilisé soit pour analyser la performance d'une installation soumise à différentes conditions opératoires, soit pour faire le dimensionnement d'une nouvelle installation. Dans ce modèle, chaque composant est considéré comme un volume de contrôle avec ses propres entrées et ses propres sorties. La performance d'un cycle est décrite par des bilans massiques sur l'eau et le LiBr, des bilans d'énergie pour chacun des composants de l'appareil et par un bilan d'énergie global et des équations de transfert de chaleur entre les flux internes et externes. Les solutions dans le générateur et l'absorbeur sont considérées comme étant à l'équilibre avec le réfrigérant sous forme vapeur à la même température et à la même pression. Des hypothèses majeures ont été faites pour simplifier la simulation et l'analyse. Celles-ci sont : une opération en régime permanent, des pertes de chaleur négligeables et des gains de chaleur entre le système et son environnement, une énergie négligeable pour le fonctionnement des pompes internes en comparaison à l'énergie alimentée au générateur et des pertes de pression négligeables dans les conduits et les différents équipements. Un algorithme typique du modèle utilisé pour simuler et analyser des PACs est présenté et montre comment les données d'entrée et de sortie peuvent changer lorsque le modèle est utilisé à des fins de conception.

Un plan expérimental a été conçu pour mesurer la performance de l'appareil, caractérisé par son COP et sa capacité de refroidissement, sous différentes conditions d'opération en utilisant comme facteurs dominants les températures d'eau chaude, d'eau réfrigérée et d'eau de refroidissement et les débits d'eau chaude et d'eau de refroidissement. Chacun de ces paramètres a été varié dans un intervalle donné par une incrémentation progressive et prédéterminée, les autres variables sont restées constantes et à leur valeur médiane. Après chaque changement de température ou de débit, le régime permanent est considéré comme atteint lorsque les valeurs des variables restent constantes pour une vingtaine de minutes.

Pour chaque groupe d'expériences, les données expérimentales et simulées sont présentées et il y a une bonne concordance entre les deux types de valeurs. Il a été montré que la pompe à chaleur a un COP élevé pour une plage étendue de températures d'entraînement, ce qui indique une utilisation efficace de la source de chaleur et en fait une bonne candidate pour la combiner avec des collecteurs solaires à plateaux et des unités de cogénération. Le COP varie très peu pour toute l'étendue des températures de l'eau réfrigérée. Cette variation a un effet plus important sur la capacité de refroidissement qui augmente de 9.2 kW à 6 °C jusqu'à 14.3 kW à 16 °C. Le niveau de température de l'eau de refroidissement a un effet plus important sur la capacité de refroidissement de la pompe à chaleur. La capacité de refroidissement augmente de façon significative alors que la température de l'eau de refroidissement diminue et le COP varie légèrement avec des températures d'eau de refroidissement se situant entre 20 °C and 32 °C. À des températures plus hautes, le COP diminue jusqu'à 0.53. Il a été montré que le débit d'eau externe au générateur a très peu d'effet sur la performance de la pompe à chaleur. Le COP est presque constant sur toute l'étendue des débits et la capacité de refroidissement est améliorée légèrement par le haut débit du générateur. La capacité de refroidissement et le COP sont améliorés par l'utilisation de hauts débits d'eau de refroidissement.

Dans la dernière partie de ce projet, il a été montré comment un modèle développé peut être combiné avec la méthodologie de mise en œuvre du dimensionnement d'une PAC sur un procédé existant. Le modèle est utilisé pour calculer les paramètres de conception et d'opération d'une PAC intégré à un procédé Kraft. Dans l'exemple sélectionné, l'objectif de la pompe à chaleur est de valoriser la charge thermique à basse température de l'effluent de la section de blanchiment et d'augmenter sa température. De l'eau chaude a été fournie à une température supérieure à la température de pincement. De la vapeur basse pression à 345 kPa a été utilisée comme énergie d’entrainement pour le générateur. A partir du modèle, les paramètres de conception de la pompe à chaleur ont été calculés en régime permanent.

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NOMENCLATURE

A	surface area (m^2)
D	tube diameter (m)
F	fouling factor (m^2°C /kW)
k	thermal conductivity (kW/m^2°C)
h	enthalpy (kJ/kg)
LMTD	log mean temperature difference (°C)
m	mass flow (L/min)
P	pressure (bar)
Q	heat capacity (kW)
T	temperature (°C or K)
U	overall heat transfer coefficient (kW/m^2-°C)
un	uncertainty
x	concentration of LiBr (%)

Abbreviations

Ab	absorber
Act	actual
AHP	absorption heat pump
CC	composite curve
Con	condenser
COP	coefficient of performance
DOE	department of energy
ECC	exergy composite curve
EGCC	exergy grand composite curve
Eva	evaporator
EXP	experimental

GCC	grand composite curve
Gen	generator
HEN	heat exchanger network
MCR	minimum cooling requirement
MHR	minimum heating requirement
OD	oven dried
PP	pinch point
P&P	pulp and paper
RTD	resistance temperature detector
SHX	solution heat exchanger
Sim	simulation
Theo	theoretical
VRHP	vapour recompression heat pump

Greek symbols

α	heat transfer coefficient (kW/ m^2-°C)
Γ	fluid mass flow rate per unit length per one side of the exchanger tube (kg/s-m)

Subscripts

a	absorbe
c	condenser
e	evaporator
eq	equilibrium
g	generator
in	inlet
out	outlet
ref	refrigerant
sat	saturation
sol	solution

CHAPTER 1 - INTRODUCTION

1.1 Justification of the project

In a global context, it is essential to reduce the fossil-based energy consumption in order to decrease the environmental impact of human activities. Industry's share of the total energy consumption is very large, and thus reduction of energy use is highly motivated in this sector.

The two problems of diminishing supplies of fossil fuels and increased CO_2 levels motivate research and development towards a sustainable energy system with less use of fossil fuels. To decrease this use, two main paths must be pursued (Nordman, 2005):

- Development and introduction of new energy production techniques, not relying on fossil fuels.
- Energy conservation (intensification) measures to use fuels more effectively.

A number of different options are possible to reduce industrial energy use. Proposed measures include e.g. heat pumping, combined heat and power generation, fuel switching, more efficient unit operations, and increased heat recovery by heat exchange.

Heat pumps (HP) are energy conversion devices that are used to upgrade the quality of heat by raising the temperature at which it is available (Herold *et al.*, 1996). Absorption heat pumps (AHP) are emerging as a potential alternative to the more common vapour recompression heat pumps (VRHP). AHPs use the effect of pressure on a solution's absorption-desorption cycle to raise the temperature of available heat to use it more advantageously at higher temperature level. They are thermally driven and when judiciously positioned into an industrial process, they can be operated with practically no purchased power. They use working fluids that are environmentally benign. However, they are more complex machines than VRHPs and require a higher initial investment but, in a context of increasing primary energy cost and greenhouse gas emission (GHG) legislations, they may represent an economically attractive approach to sustainable development.

Pinch Analysis is a technique used to maximize internal heat recovery within a process (Linnhoff, 1993; Noel and Boisvert, 1998). It has been applied to a broad spectrum of industries including the P&P industry (Rouzinou *et al.*, 2003; Savulescu *et al.*, 2005). The thermodynamics of Pinch Analysis dictates a fundamental rule: there must not be transfer of heat from above to below the pinch point. If this happens, the process suffers a double penalty: the simultaneous increase of the cooling and heating requirements of the process. On the other hand, an HP must transfer heat in the opposite direction from below to above the pinch point; so it should be integrated in such a way that the heat source is situated where there is an excess of heat, i.e. below the pinch, and the heat sink where there is a need for heat, i.e. above the pinch. The methodology of integration of traditional heat pumps such as vapour recompression heat pumps or electrically driven compression heat pumps is well known and discussed in the literatures (Wallin *et al.*, 1990; Wallin and Berntsson, 1994). However integration of more complex configurations like absorption heat pumps and absorption heat transformers in a process has not been thoroughly investigated.

The pulp and paper industry (P&P) is Canada's premier industry characterized by very large energy requirements in the form of electricity and thermal energy (27% of all industrial energy consumption (Browne *et al.*, 2006)). An old kraft mill consumes an average of 25 GJ/(t of oven dried (OD) pulp produced), while a modern mill consumes 12 GJ/t (Browne, 1999) of thermal energy. Energy costs account for one third of manufacturing costs in Canadian mills and competition in the pulp and paper industry has become fiercer over the past few years; high energy consumption has, therefore, become an issue. Also, Canada is one of the countries, that signed the Kyoto protocol with the aim of reducing green house gas emissions by 6% below the value of 1990 (Browne, 2003). For these reasons, the P&P industry must enhance its efforts to improve energy efficiency and, subsequently, thermal energy consumption and gas emissions. It seems that implementation of absorption heat pumps is an appropriate way to reach these aims.

1.2 Context of the Project

This research is part of the E^3PAP project, which is a feasibility study on novel technologies for energy efficiency and eco-industrial clusters in the pulp and paper industry. It is executed at École Polytechnique de Montréal under the co-ordination of Professor Jean Paris. Its objective is to ascertain whether or not it is technically feasible and cost effective to integrate Canadian pulp and paper mills into eco-industrial clusters by retrofitting advanced thermal cycles for enhanced energy efficiency. The focus is on two areas: the mill supplying energy and the surrounding community using it.

A Kraft mill located in the eastern Canada and the small town beside (population = 2500 habitants) are the subject of the case study.

The E^3PAP project consists of three parts:

Part 1: An in-depth energy analysis of the Mill. This part will focus on developing technically feasible solutions to reduce the energy requirements of the mill, thus creating a surplus of energy that can be sold to the town.

Part 2: A web-based community survey, analyzing the heating demands of the community surrounding the mill. The objective of this part is to characterize and quantify the potential uses of exportable energy (power, hot water, low pressure steam) produced by the mill.

Part 3: A feasibility study of implementation of absorption heat pumps in the process based on the in-depth energy analysis of the mill and results of Pinch Analysis.

This PhD project is completed in the core of the last part of the E^3PAP project, the process energy analysis part and implementation of AHPs in the process.

1.3 Heat Pumps

Heat pumps (HP) are thermal machines used to increase the temperature at which a certain amount of heat is available; this temperature lift is a key characteristic of a HP. This transfer can only be accomplished with an input of energy. For the more common vapour recompression heat pumps (VRHP), this energy is supplied to the system as mechanical work. One example of a work driven heat pump is the common refrigerator used in most households. The component that requires an energy input is a mechanical compressor and is typically driven by an electric motor.

In this section the principles of HPs and AHPs briefly will be described.

1.3.1 Compression Heat Pumps

The vapour recompression cycle is the cycle most often used for air-conditioning and refrigeration applications. Figure 1-1 shows a schematic of the equipment layout for a typical VRHP cycle. It consists of four main components: compressor, condenser, evaporator and expansion valve, which are connected by a piping system (Dorgan *et al.*, 1995).

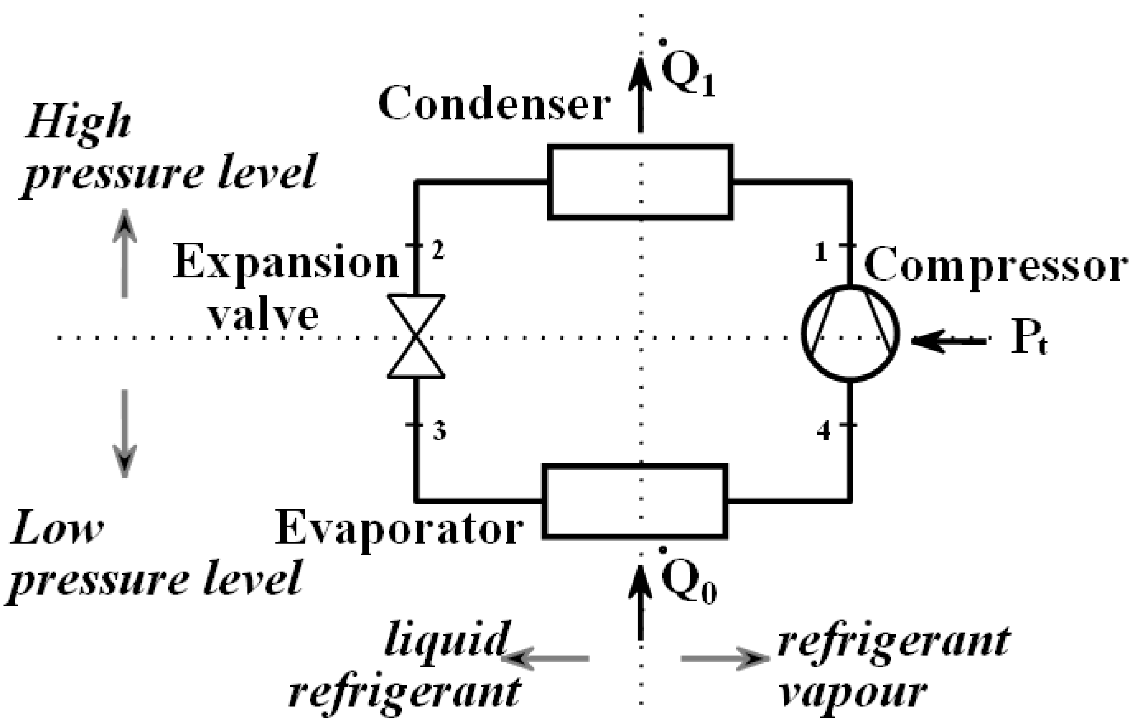

Figure 1-1 Schematic of equipment layout for typical vapour compression cycle

Superheated vapour leaves the compressor (1) and is liquefied in a condenser by heat exchange with a cooling fluid (usually water or air). The liquid refrigerant (2) then passes through an expansion valve. The expanded liquid (3) enters the evaporator where it absorbs heat from the medium to be cooled and is vaporized. The vapour (4) then enters the compressor and is raised to a higher pressure level (Dorgan et al., 1995). Figure 1-2 shows the cycle in a pressure-enthalpy-diagram.

While a classification of vapour-compression systems based on their driving force (e.g. electrical motor, gas and diesel engines, gas and steam turbines) is possible, they are most often categorized based on their type of compression device:

- Reciprocating compressor
- Centrifugal compressor
- Screw compressor

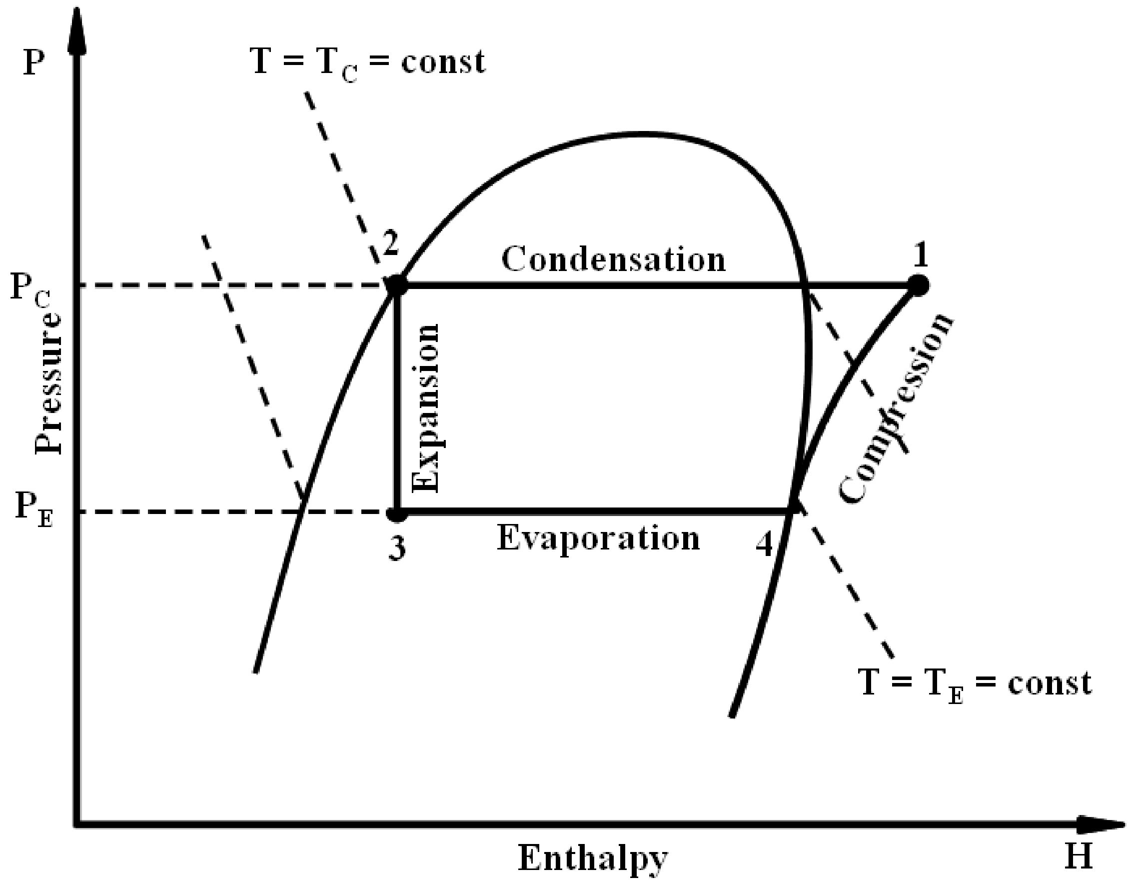

Figure 1-2 Pressure-enthalpy-diagram of conventional vapour-compression cycle

Referring to temperatures and heat flows indicated in Figure 1-1 and Figure 1-2 the following parameters can be defined:

Temperature lift:

$$\Delta T_{lift} = T_C - T_E \tag{1-1}$$

The temperature lift of vapour compression heat pumps is defined as the difference of temperature of heat consumer T_C and heat source T_E. It is defined here for later comparison to the lift of absorption heat pumps.

Coefficient of performance:

$$COP = \frac{heat \quad output}{work \quad input} \tag{1-2}$$

Depending on whether the cycle is used for cooling or heating purposes, the heat flow to be defined as useful heat output varies: for chiller applications it is $\dot{Q}_0$, and for heating applications it is $\dot{Q}_1$. The input work is P_t.

Carnot efficiency:

The Carnot efficiency depends only on the involved temperature levels of heat transferred to and from the system and is independent of working fluids used or kind of machinery or process it is applied to (Sonntag *et al.*, 2003). It is therefore commonly used as a point of comparison of thermodynamic processes.

$$COP_{cooling} = \frac{T_E}{T_E - T_C} \tag{1-3}$$

$$COP_{heating} = \frac{T_C}{T_E - T_C} \tag{1-4}$$

Based on the Carnot efficiency a so called Carnot rating can be defined:

Carnot rating:

$$\eta_C = \frac{COP}{COP_{Carnot}} \tag{1-5}$$

1.3.2 Absorption Heat Pumps

An absorption system was first developed by John Leslie in Scotland which was operated intermittently. The first continuously operating NH_3/H_2O was built by Ferdinand Carre in 1859 in France. The first commercialized absorption system operating with $LiBr/H_2O$ was manufactured by the U.S. Carrier Company in 1945. A discussion of the chemical background of absorption, fundamental of absorption heat pumps, different working fluid pairs and multistaging is provided below.

1.3.2.1 Chemical Background of Absorption

Absorption, in chemistry, is a physical or chemical phenomenon or a process in which atoms, molecules, or ions enter some bulk phase - gas, liquid or solid material. It is contrasted with adsorption, in which the molecules adhere only to the surface of the second substance. Physical absorption involvs factors such as solubility and vapor-pressure relationships. Chemical absorption is involving chemical reactions between the absorbed substance and the absorbing medium. Two chemical pairs are generally used for the absorption process. The water - ammonium pair is used for air conditioning and cooling. In that case, the cooling agent is the ammonium while the absorbent is the water. The reaction of ammonium and water is an exothermic reaction (Ziegler and Riesch, 1993).

$$NH_3 + H_2O \rightarrow NH_4OH + \text{energy} \tag{1-6}$$

The other pair is lithium bromide - water. That pair is used for every cases where evaporating temperatures above 0 ˚C are needed. In that pair the water is the cooling agent and the lithium bromide is the absorbent.

$$LiBr + H_2O \rightarrow HBr + LiOH + energy \quad (1\text{-}7)$$

1.3.2.2 Fundamental of Absorption Heat Pumps

Absorption systems use latent heat of the liquid-vapour phase change and take advantage of the increased boiling point of solutions compared to pure liquids. Thus sorption systems use two working fluids: the refrigerant and the sorbent. The phase transition temperature can therefore be manipulated not only by changing the pressure (as done in vapour compression heat pumps) but also by changing the refrigerant concentration (Ziegler and Riesch, 1993) in the sorbent.

The key difference between the absorption cycle and the vapour-compression cycle is the process by which the low-pressure refrigerant vapour is compressed to a high pressure. Figure 1-3 shows the equipment layout of an absorption cycle in a one-stage water/LiBr absorption chiller. Similar to the vapour-compression cycle, in absorption cycle the refrigerant is condensed, throttled and finally vaporized in the evaporator. However, instead of compressing this vapour using mechanical energy, it is absorbed into a solution to be transported as a liquid by a pump to the high pressure level, where the refrigerant vapour is again driven out of the solution by heat input. By replacing the mechanical compressor of the vapour compression cycle with this “thermo chemical compressor” consisting of absorber, generator, solution pump and expansion valve, the amount of work required for the cycle is reduced considerably: as the specific volume of a liquid is by far smaller than that of a gas, the work for the solution pump in an absorption system is much smaller than that of the compressor in a conventional vapour compression heat pump (equation 1-8).

Work:

$$w_t = \int v \cdot dp \tag{1-8}$$

The schematic of Figure 1-3 is drawn as if superimposed on a van t'Hoff diagram of the working fluid pair. Such a diagram depicts the logarithmic vapour pressure as a function of the temperature ($-1/T$). The relative position of the components with phase change in the schematic therefore indicates the relative temperature and pressure of the working fluid inside those components. The exception to this are the subcooled and superheated states, which cannot be accurately represented in a van t'Hoff diagram that only displays saturated states (Herold et al., 1996).

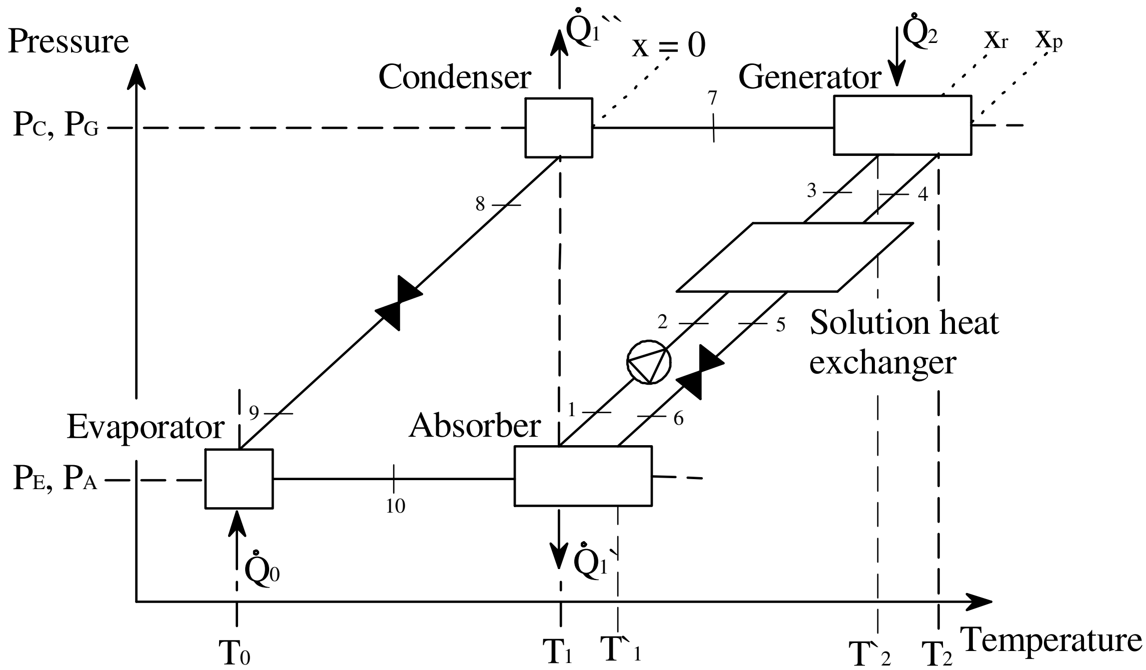

Figure 1-3 Schematic of one-stage absorption heat pump

The advantage of this representation is that the isosteres, vapour pressure curves of solution of constant concentration, are close to straight lines. In Figure 1-4 a van t'Hoff diagram is shown for the working fluid pair water/lithium-bromide. The solution field is limited on the low temperature side by the water line (concentration x = 0), while the limit on the high temperature side is represented by the 100% sorbent concentration line (e.g. NH_3/water) or, as in the case of Lithium/Bromide, the crystallization line. The figure clearly demonstrates the increase of the boiling point of the solution with growing concentration of the refrigerant.

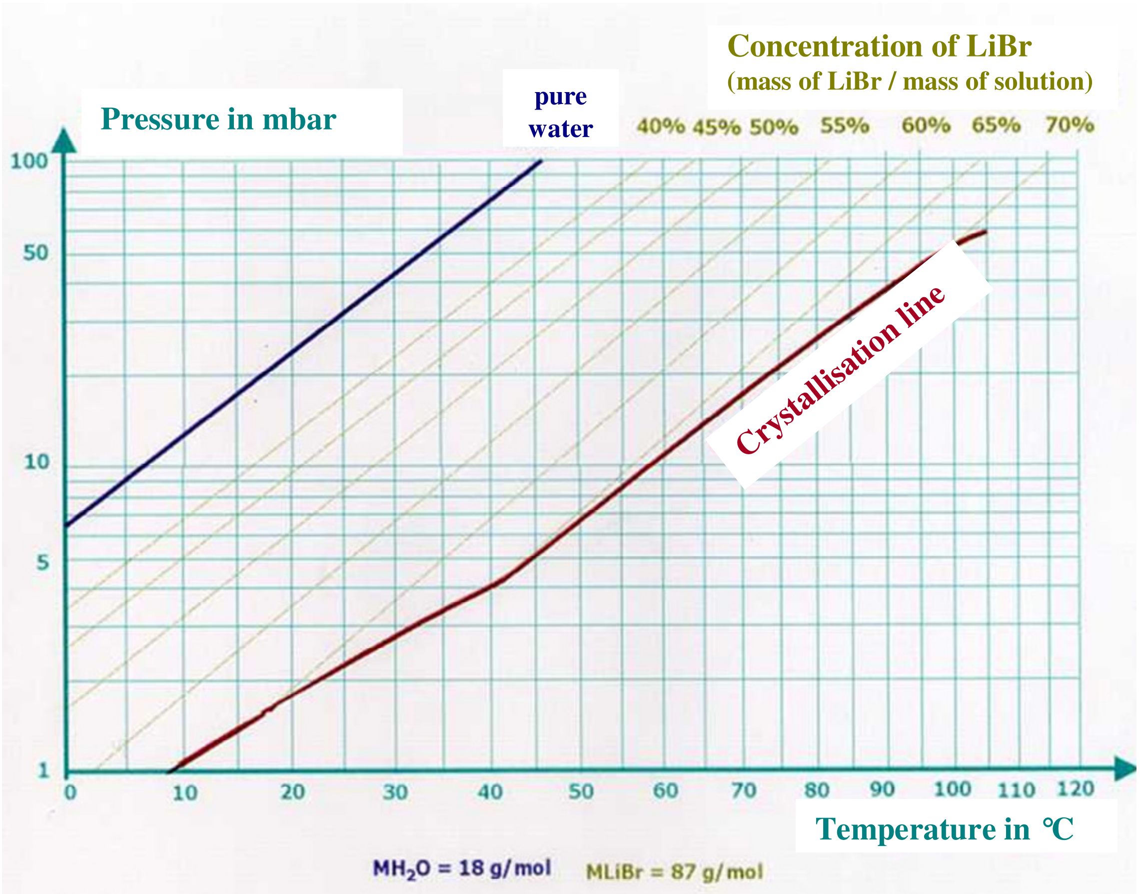

Figure 1-4 Van t'Hoff diagram for solution of water/lithium bromide

A variety of different types of absorption heat pumps exist, differing according to working fluid pairs, application guided design, etc. In principle however, they are all based on the same thermodynamic cycle. Main components of this cycle are an evaporator, an absorber, a generator and a condenser, as shown in Figure 1-3.

A diluted low-pressure solution, rich in refrigerant (see Figure 1-3), (1) is pumped to a higher pressure level (2) and preheated in a solution heat exchanger (3). It reaches the generator where refrigerant (7) is boiled out of the solution by heat input. While the remaining concentrated solution (poor in refrigerant) passes through the solution heat exchanger (5), is expanded to the low-pressure level (6) and returned to the absorber, the refrigerant vapour (7) flows to a condenser, where it is liquefied under heat rejection (8). Pressure is then reduced to the low-pressure level (9) before the refrigerant reaches the evaporator, where it is again evaporated under heat input (10). It then flows back to the absorber, where it is absorbed into the concentrated solution (1). The task of the solution heat exchanger is to block the sensible heat in the generator (a part of the used driving heat) avoiding its transport into the absorber, where it would be discharged to the environment, which, in turn, would increase the cycle losses and deteriorate the heat pump COP creating a short circuit between driving and waste heat. This configuration of an AHP is also referred to as an AHP type I.

Apart from classifying absorption systems according to the working fluid used or the stages involved, they are usually classified according to the type of heat input. Absorption machines using steam, hot water or hot exhaust gases as heat source are referred to as indirect-fired machines, while those, which directly burn gas and have their own heat source, are called direct-fired machines. Standard absorption units are available in the range of some hundred kW to some MW capacity. They are normally used as cooling machines for air-conditioning purposes. Direct-fired or steam driven double-effect (see section 1.3.2.6) and single-effect chillers driven with heat of about 80-160°C[1] are manufactured by various companies, located predominantly in the Far-East and the United States. Recently, chillers with lower capacities (50 to 200 kW) have been marketed for trigeneration systems. However, presently small scale indirect fired absorption chillers below 35 kW cooling capacity are not commercially available, thus leaving a gap for small scale applications like solar cooling systems, micro-cogeneration and trigeneration plants. The coefficient of performance of absorption heat pumps lies between 0.5 and 1.2, depending on

[1] single effect 80-110°C, double effect 130-160°C

the cycle used (Herold et al., 1996; Srikhirin *et al.*, 2001; Ziegler and Riesch, 1993). Absorption heat pumps generally use a mechanical pump to circulate the liquid sorbent solution in the chillers. The electrical input is normally in the range of 1 to 5% of the cooling capacity.

1.3.2.3 Working Fluid Pairs

The performance of a real heat pump is lower than that of the ideal heat pump, mostly due to irreversibilities caused by technical reasons like heat or pressure losses. Nevertheless the thermodynamic properties of the working fluids involved also play a role (Chen and Wu, 2001). A suitable refrigerant for an absorption system has the following characteristics (Hainsworth, 1944; Yakubov *et al., 1994*):

- Mass and heat transfer properties satisfactory over the operating temperature range
- High solubility in sorbent solution at the absorber operating temperature
- Low solubility in sorbent solution at the generator operating temperature
- No irreversible reactions with sorbent within operating temperature range
- Sorbent should have a low vapour pressure compared with the refrigerant
- Sorbent should have a low heat capacity

Technical requirements must be additionally met by the working solution (refrigerant and sorbent):

- Non-corrosive to equipment within operating temperature range
- Non-toxic

- Harmless to the environment
- Non-flammable and non-explosive
- Inexpensive

The working fluid pairs commonly used in commercial absorption heat pumps up to today are water/LiBr and ammonia/water. Many other working fluids have been considered for absorption machines, such as water/sodium hydroxide, water/sulfuric acid, ammonia/sodium thioczanate, and others (Hainsworth, 1944; Sozen *et al.*). However, none of these alternatives has been able to establish any market foothold.

In the case of the water/LiBr working pair the salt solution serves as absorbent and water as the refrigerant limiting the lowest temperature of the cycle, T_0, to a temperature above the freezing point of water. These cycles are therefore usually used as chillers for air-conditioning purposes. Ammonia/water cycles, using ammonia as refrigerant and therefore able to reach temperatures below zero, are used for refrigeration applications. As can be seen in Figure 1-4, the solution field of water/LiBr is also limited towards high temperatures by the crystallization line. At concentrations or temperatures passing this line, the salt in the solution suddenly begins to crystallize and fall out of the solution, stopping the functionality of the unit. This restriction does not apply for NH_3/water (see appendix A-1 for respective van t'Hoff diagram), which can consequently achieve much higher temperature lifts and is therefore also suitable for heating purposes (heat pump mode).

Another important difference is the vapour pressure of the two refrigerants. While ammonia has a relatively high vapour pressure at the temperatures required for common applications (usually pressures in the range of 20-30 bar), that of water is lower than atmospheric pressure (5-20 mbar). This obviously has major impact on the construction and design of the main heat exchangers and the connecting piping and equipment.

It should also be noted, that for the working fluid pair ammonia/water the vapour pressure of the absorbent, water, is in the same range as that of ammonia. As a consequence, the vapour generated in the generator contains a certain amount of water that must be rectified to maintain performance at an acceptable COP.

The properties of the working fluid pairs ammonia/water and water/lithium bromide are summarized in Table 1-1.

Table 1-1 Absorption working fluid properties (Herold K. E. et al., 1996)

Property	**Ammonia/Water**	**Water/Lithium Bromide**
Refrigerant	Ammonia	Water
high latent heat	good	excellent
vapour pressure	too high	too low
low freezing temperature	excellent	limited application
low viscosity	good	good
Absorbent	Water	Lithium Bromide
low vapour pressure	poor	excellent
low viscosity	good	good
Mixture		
no solid phase	excellent	limited application
low toxicity	poor	good
high affinity between refrigerant and absorbent	good	good

1.3.2.4 Definition of the parameters

Next the thermodynamic parameters of a sorption heat pump will be discussed. The parameters have been defined in Figure 1-3.

Temperature lift:

Analog to compression chillers, the temperature lift of absorption chillers is defined as the difference of temperature between heat sink T_1 and heat source T_0.

$$\Delta T_{lift} = T_1 - T_0 \tag{1-9}$$

Temperature thrust:

The definition of temperature thrust is the difference of driving heat at a temperature T_2 to that of the heat consumer T_1.

$$\Delta T_{thrust} = T_2 - T_1 \tag{1-10}$$

Coefficient of performance:

The general definition for the COP is:

$$COP = \frac{heat\ output}{heat\ input} \tag{1-11}$$

Depending on whether the cycle is working in cooling or heating mode, as a single- or a multi-effect machine, a heat transformer or a heat pump, the heat flows to be defined as useful heat output and driving heat input vary; in the case of a single-stage chiller and referring to temperatures and heat flows indicated in Figure 1-3, the definition of COP is the following:

$$COP = \frac{\dot{Q}_0}{\dot{Q}_2} \tag{1-12}$$

In order to get more information about the thermodynamic potential of the cycle it is useful to discuss the reversible limit of the COP, which is well-known and proportional to the ratio of temperature thrust and temperature lift (Ziegler, 1999).

$$COP_{rev} = \frac{T_0 \cdot (T_2 - T_1)}{T_2 \cdot (T_1 - T_0)} \sim \frac{\Delta T_{thrust}}{\Delta T_{lift}} \geq COP \tag{1-13}$$

Concentration:

The concentration of a working solution in an absorption heat pump is defined as the ratio of mass of absorbent to mass of solution.

$$x = \frac{m_{absorbent}}{m_{solution}} \tag{1-14}$$

with:

$m_{absorbent}$ = mass of absorbent $\qquad$ $m_{solution}$ = mass of solution

The concentrations of poor and rich solution in an absorption heat pump are dependent variables. Equations (1-15) and (1-16) present the mass balance equations for a single stage AHP, where the indexes *r*, *p* and *vap* stand for rich, poor and vapour.

$$\dot{m}_r = \dot{m}_p + \dot{m}_{vap} \tag{1-15}$$

$$x_r \cdot \dot{m}_r = x_p \cdot \dot{m}_p \tag{1-16}$$

Solution recirculation rate:

The recirculation rate f is defined as the ratio between the solution flow pumped from the absorber into the generator and the vapour flow produced in the evaporator. The higher f, the more solution is circulated between absorber and generator for a given cooling capacity.

$$f = \frac{\dot{m}_r}{\dot{m}_{vap}} \tag{1-17}$$

with:

$\dot{m}_r$ = mass flow of rich solution $\dot{m}_{vap}$ = mass flow of water vapour

1.3.2.5 Absorption Heat Transformer

In an absorption heat transformer (AHT or AHP type II), the high and low pressure zones of the machine are inverted (Figure 1-5). The evaporator and absorber operate at the higher pressure and the condenser and generator at the lower pressure (Costa *et al.*, 2009). This also changes the temperature levels of the various heat exchangers so that the driving heat is now supplied to the evaporator-generator pair at the intermediate temperature level while useful heat is released by the evaporator at high temperature and by the condenser at low temperature. The absorption heat transformer offers interesting possibilities in a process integration context when there is a need for low temperature heat (for example to preheat fresh water).

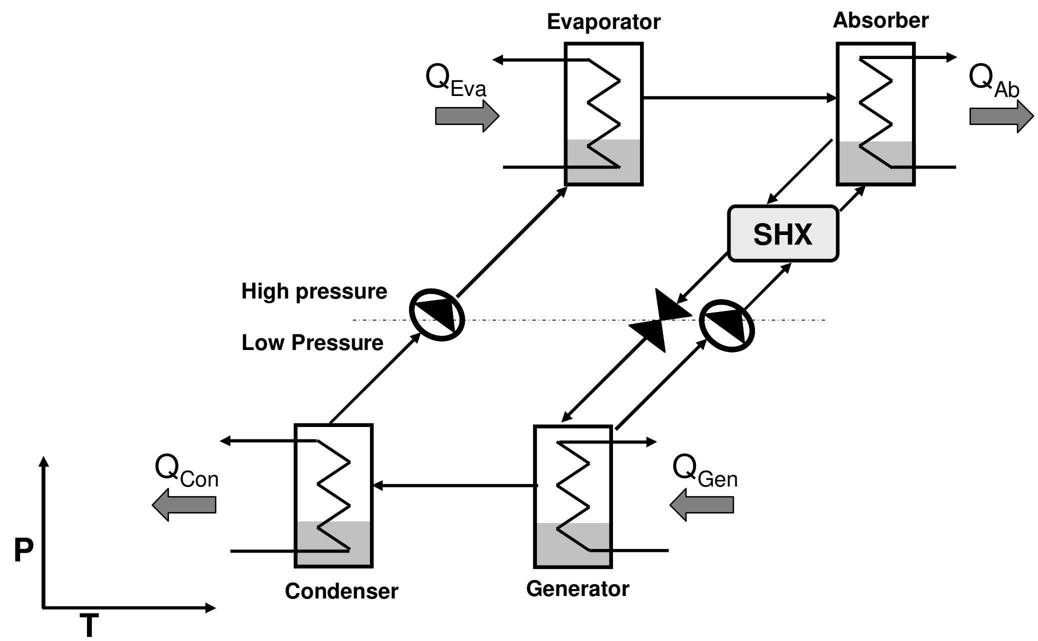

Figure 1-5 Schematic of an AHT (AHP type II)

1.3.2.6 Multistaging

The single-effect cycles are limited in their use by the geometry of the solution fields. This is especially drastic in the case of water/lithium bromide heat pumps where the risk of crystallization increases with high solution concentration and low temperatures. At low temperatures, the freezing point of the refrigerant becomes the limiting factor. These limitations however can be surmounted by multistaging. The basic idea is to repeat either the desorption-condensation processes or the evaporation-absorption processes at different pressures or temperatures as compared to the single effect cycle (Ziegler, 1999), thus reaching higher temperature lifts (double-lift) or higher efficiencies (double-effect).

Three simple examples are shown in **Error! Reference source not found.**, which is a pressure-temperature plot with the solution field and the axes not shown. The circles represent the main exchangers, the arrows denote the heat flowing in or out of a component. Consequently, circles with arrows entering represent evaporators or generators, circles with arrows leaving represent absorbers or condensers. The number of arrows qualitatively represents the respective heat load.

In **Error! Reference source not found.**a heat of condensation is used internally to regenerate the solution for the second time in a middle temperature generator. In **Error! Reference source not found.**b the same is accomplished using the heat of an absorber. In **Error! Reference source not found.**c heat of low pressure absorber is used internally in a middle pressure evaporator. The first two cycles are of the double-effect type, whereas the third cycle is of the double lift type (Ziegler, 1999).

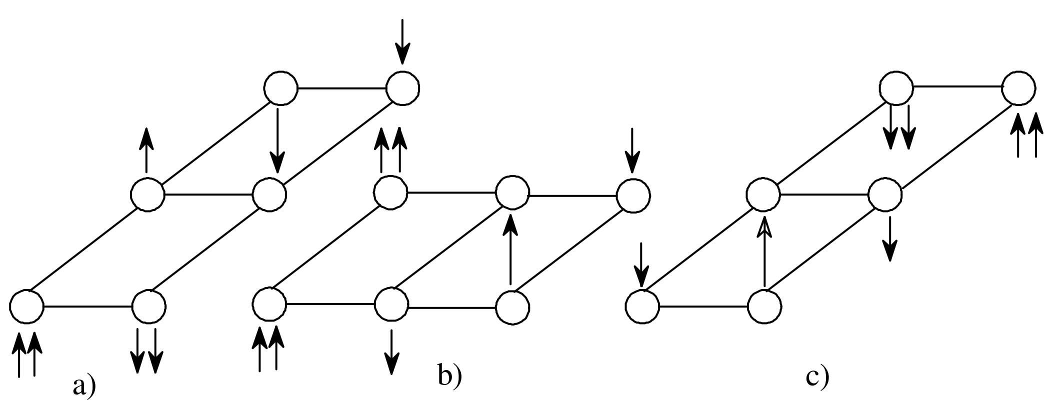

Figure 1-6 Drawing of multi-effect cycles (a and b double effect, c double lift) in the pressure-temperature plot. Arrows denote heat flow (Ziegler, 1999)

1.4 General Objective

As discussed in the previous sections, there is a lack of a methodology for positioning and dimensioning absorption heat pumps (AHP) implemented in a process, and this is the main purpose of this project. Also it will be determined whether it is technically feasible and cost effective to improve energy efficiency of a Kraft mill by the retrofit installation of AHP in the process on the basis of the developed methodology.

1.5 Organization of the Thesis

Chapter I introduced the general objective of this research. It also provides background for this project, as well as the principles and operation of heat pumps. The thermodynamic parameters of a sorption heat pump are discussed; vapour compression and absorption heat pumps are compared and the characteristics of suitable refrigerants for an absorption system are introduced.

In Chapter II, earlier methods presented for heat pump integration and previously obtained results are discussed. Also a review of different implementations of HPs in the pulp and paper industry is presented and different available models for AHPs and experimental work are described. In the last section an analysis of the literature allows for the determination of the specific objectives.

In Chapter III, overall methodological approach of this research and organization of the submitted articles are presented.

In Chapter IV, the first article is presented; retrofit of absorption heat pumps into manufacturing processes: a general methodology.

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In Chapter V, the second article is presented; opportunities for the integration of absorption heat pumps in the pulp and paper process.

In Chapter VI, the third article is presented; simulation and dimensioning of H_2O-LiBr absorption heat pumps.

Finally, a general discussion and conclusions are presented in Chapter VII, while recommendations for future research are stated in Chapter VIII.

CHAPTER 2 - LITERATURE REVIEW

In this chapter, earlier methods presented for heat pump integration and previously obtained results will be discussed. The aim is to outline the field of heat pump integration and the background for this work. The presentation below does not include all published works; only the most relevant ones have been selected. In this literature review I have also referred to some PhD theses (Abrahamsson, 1993; Kohlenbach, 2005; Lee, 1999; Nordman, 2005; Schaefer, 2000; Wallin, 1996).

In the first section different methods for analyzing heat pump integration are discussed. In the second section, some larger studies on the potential for heat pump integration are reviewed and different implementations of HPs in the pulp and paper industry are analyzed. In the last section, some available models of AHP and previous experimental works are briefly discussed with their advantages and drawbacks.

2.1 Methods for Process Integration of Heat Pumps

2.1.1 Earliest Integration Methods

The first methods used a unit operation-oriented approach to identify heat pumping opportunities. In that approach, heat pumps were integrated into the chosen large utility consumers, such as distillation columns and evaporators. A typical example (Figure 2-1) is the insertion of a heat pump between the condenser and the reboiler of a distillation column. This method is used in several separations such as: isobutene/n-butane (Barnwell and Morris, 1982), propane/propylene (Quadri, 1981a, 1981b), ethane/ethylene (Menzies and Johnson, 1971).

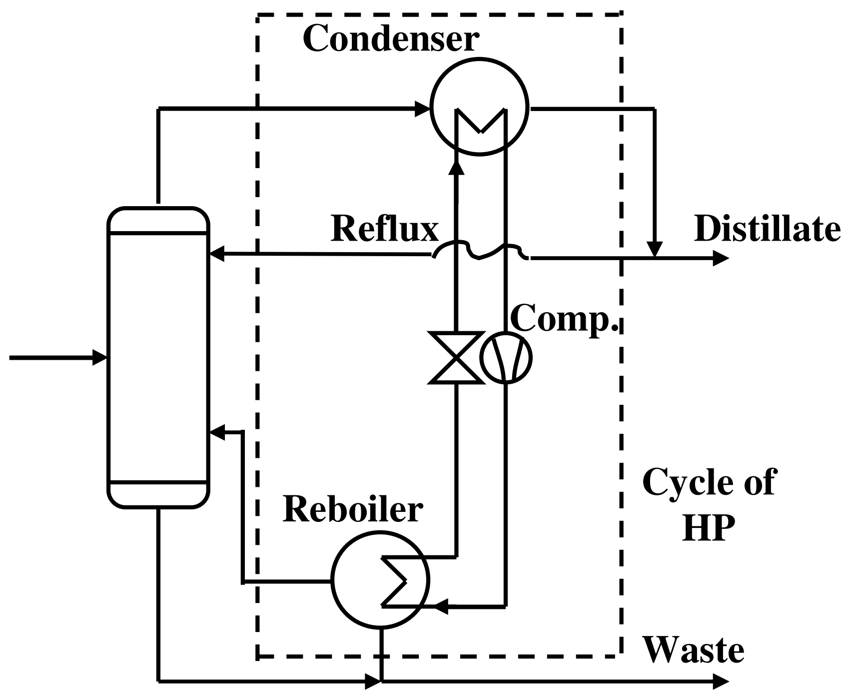

Figure 2-1 Installation of an heat pump in a distillation unit

Barnwell and Morris (1982) showed how energy use can be cut by nearly 50% by using a heat pump installation in operation of a butane splitter. It was shown that compared to a conventional fractionator, the heat pump increases the capital cost and the payback time. This increase in payback time was calculated at less than one year.

Quadri (1981a; 1981b) presented the process design aspects of thermocompression (heat pump) rectification, with particular reference to propylene-propane (P-P). The basic configuration and optimization problems are both considered. In Part I, alternative process schemes and a computer oriented calculation method for design optimization of heat pump splitters are presented. Part II, discusses the results of optimization obtained for applying the method to an industrial P-P splitter. It was concluded that the heat pump rectification is the preferred solution for P-P splitters. It was also mentioned that conventional rectification is likely to be confined to low capacity splitters, where the operating cost savings obtained by heat pump are not sufficient to overcome the higher investment required.

Menzies and Johnson (1971) have modeled a vapor recompression ethane/ ethylene distillation section of the Light Ends Recovery Unit at the Polymer Corporation, Sarnia, Ontario, using the CHESS program (Motard *et al.*, 1968). CHESS is a modular, steady state simulation system developed by the University of Houston. They performed a parametric study with the model, both to find the range of applicability of the model and to effect improvements in the unit operation.

Supranto et al. (1986) have experimentally studied the operating characteristics of a heat pump assisted distillation system using R11 as the external working fluid. The heat pump working fluid extracts heat from the top of the column, increases the temperature of the recovered heat and recycles it to provide the input heat in the reboiler. The interaction between the external parameters and the internal parameters for a specially designed heat pump assisted distillation system has been presented. The external parameters were temperature, concentration and mass flow rate of the feed, the concentration of the top and bottom products and the mass flow rate of the working fluid. The internal parameters were the energy consumption, the COP, the temperatures at the top and bottom of the column and the condensation and evaporation temperatures. The mass flow rate of the working fluid has found to be the parameter which has the greatest effect on the performance of the system. They have shown that the coefficient of performance (COP) and the temperature lift ($T_{Condenser} - T_{Evaporator}$) can be related by equation 2-1.

$$(COP)^{-1} = 0.00702(T_{Condenser} - T_{Evaporator}) - 0.0905 \quad (2\text{-}1)$$

Ferre et al (1985) presented the results of a simulation and optimization computer program to analyze the economical profitability of substituting the conventional reboiler and condenser of an existing distillation column with a direct vapor recompression heat pump. Two different cases were presented, which corresponded to the separation of organic compounds of medium molecular weight of close boiling points. The payback time for substituting the conventional distillation column by an overhead vapor recompression heat pump, was calculated at about 2.5 years for these kinds of separations. It was shown that the value of the payback time is very sensitive to the variation of the column capacity, the ratio of energy prices/ equipment costs, and the relation between cooling water and electricity cost, and it is necessary to take into account the possible re-using of the conventional equipment to be substituted.

Eisa et al. (1987) presented a feasibility study on matching a heat driven absorption heat pump to a distillation process. It was shown that the performance of an absorption heat pump is a function of the temperatures in the evaporator, the condenser, the absorber and the generator and the ratio of the mass flow rate in the secondary circuit to the mass flow rate in the primary circuit. It was presented that in absorption systems design choices are limited by the Gibbs phase rule. Plots were given of the coefficient of performance against the temperatures of the top and bottom products and also against the energy saved.

Flores et al. (1984) presented a comparison between vapor recompression and conventional designs for distillation columns and, it was shown that vapor recompression generally allows for best return of additional investment. The optimum design and operating conditions of a propane-propylene splitter using a direct vapor recompression system was presented. That system was selected for that study, since both components have similar volatility and the separation requires a column with a high number of stages operating with a high reflux ratio.

2.1.2 Integration Methods Based on Pinch Analysis

The Pinch Analysis has been used in several works to identify suitable heat pump installations in industrial processes. Pinch Analysis is a technique used to maximize internal heat recovery within a process (Linnhoff, 1993; Noel and Boisvert, 1998). Its scientific principles and utilization procedures have been described in engineering manuals (Labidi J. *et al.*, 1999; Linnhoff *et al.*, 1994). A basic step of the method is the representation in the temperature vs. enthalpy diagram of the aggregate of all possible heat transfers between process streams (Figure 2-2). It consists of two composite curves (CC), one that represents the streams which can be used as heat sources (hot streams) and one the streams which can be used as heat sinks (cold streams). The development of the composite curves, their relation in the T vs. ΔH graph and their significance have been described elsewhere (Linnhoff et al., 1994). From the CCs, the minimum heating requirement (MHR) of the process, minimum cooling requirement (MCR) of the process and the process pinch point can be identified. Another basic tool of heat exchanger network design is the grand composite curve (GCC), for which an example is Figure 2-3. The GCC presents as a temperature enthalpy profile, the external heating and cooling requirements after heat recovery has taken place. It gives a clear picture of the interface between the process and the utility system. It allows an appropriate mix of heating and cooling utilities to be selected before design.

The grand composite curve can be derived from either the composite curves or the process streams Table.

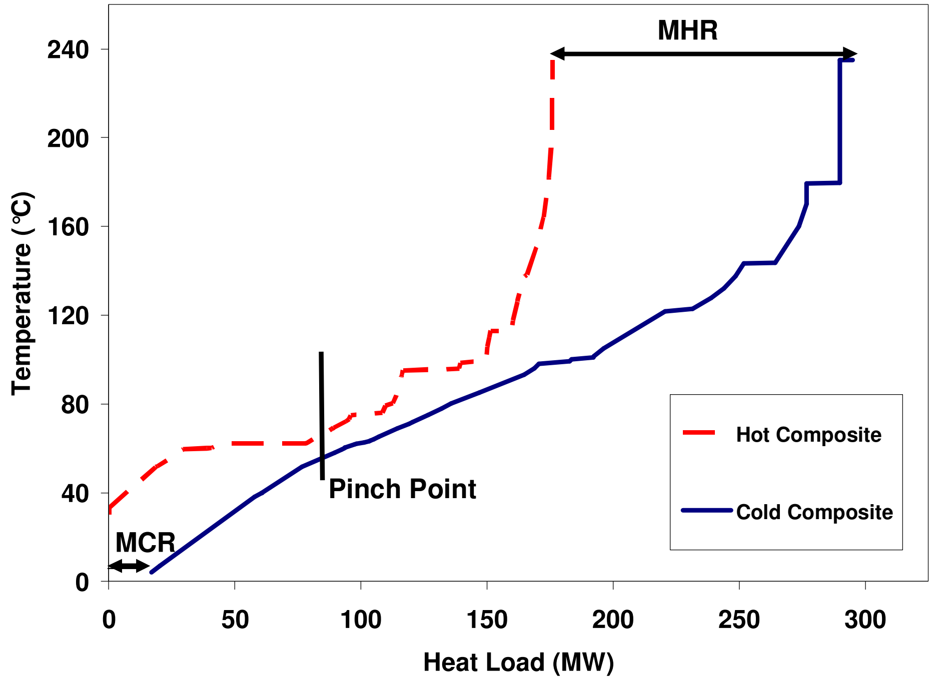

Figure 2-2 Example of Composite Curves

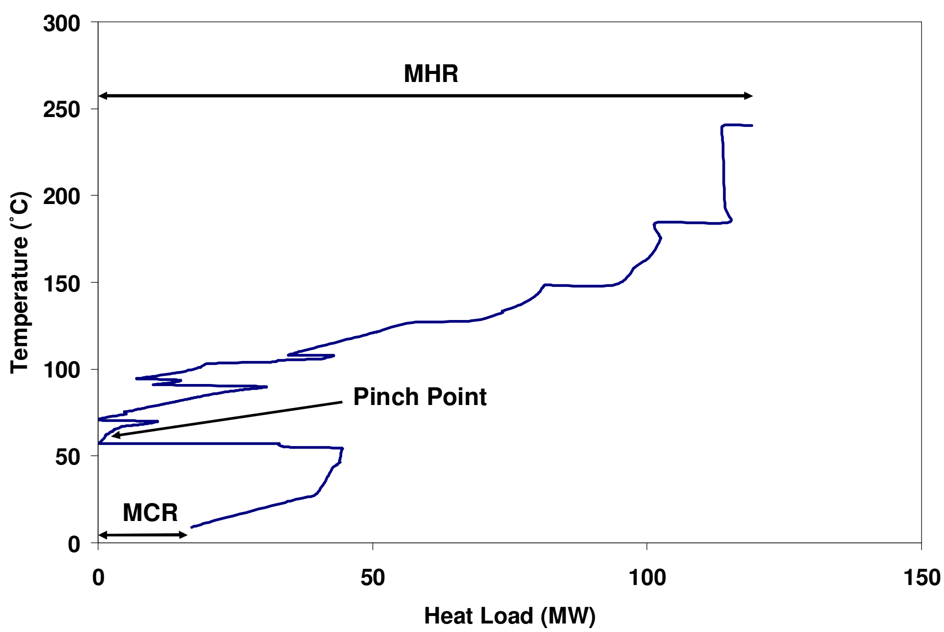

Figure 2-3 Example of Grand Composite Curve

The thermodynamics of Pinch Analysis dictates a fundamental rule: there must not be transfer of heat from above to below the pinch point. If this happens, the process suffers a double penalty: the simultaneous increase of the cooling and heating requirements of the process. On the other hand, a HP must transfer heat in the opposite direction from below to above the pinch point; so it should be integrated in such a way that the heat source is situated where there is an excess of heat, i.e. below the pinch, and the heat sink where there is a need for heat, i.e. above the pinch. In Figure 2-4, this statement is shown on a composite curves diagram with a VRHP. This section briefly summarizes some of the developed methods of positioning heat pumps using process integration and Pinch Analysis.

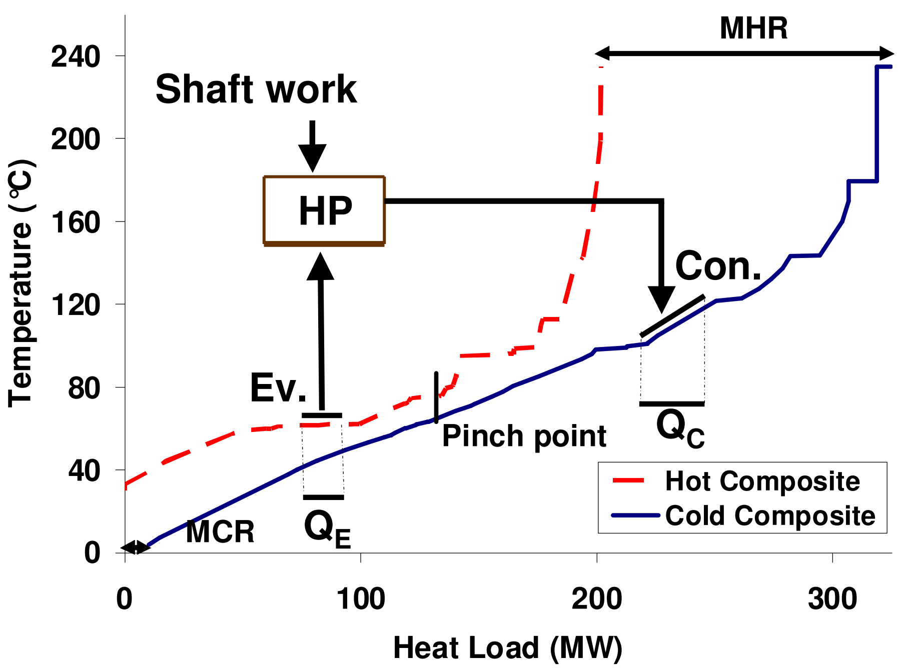

Figure 2-4 Appropriate positioning of an HP

Townsend and Linnhoff (1983) presented a method for the integration of cogeneration systems and HPs in the process. It was shown that for a cogeneration plant, the maximum amount of work is obtained by matching the temperature profiles of the cogeneration plant against the temperature profiles of the process. The integration of refrigeration systems was discussed and it was shown that such a method can also be used for analyzing heat pumps working with non-azeotropic mixtures. They recommended that economizer-couplings should be used to minimize the shaft work. The shaft work is reduced by choosing the refrigeration temperatures in a way that a better fit between them and the GCC is obtained. Since the analysis was only based on the process stream data and no economic assessments were made, the method can be used for estimating the thermodynamic potential of the techniques.

Townsend and Linnhoff (1984) presented a method to calculate the minimum required heat exchanger area as a function of the composite curves and the minimum approach temperature (ΔT_{min}). Using that method, new heat exchanger area could be calculated when a HP is implemented. Based on that method, Ranade (1988) and Ranade and Sullivan (1988) presented a method for heat pump integration, considering additional heat exchanger area, the other cost and savings as functions of heat pump size, working temperatures and global ΔT_{min} . Depending on the economic criteria used, the equations developed could be used to identify the temperature lift at which the longest acceptable pay-back period is obtained, or to compute the pay-back period, investment cost and annual savings for a number of heat pump sizes integrated at different condenser and evaporator temperatures. For a heat pump operating in a temperature region bounded by the temperatures of a low and high level utility (illustrated in Figure 2-5) with marginal values $(MV)_c$ and $(MV)_h$, a general equation was presented to obtain the maximum economic lift (for a certain period) as follows:

$$T_A / T_D = \{[C_{HPE} + C_{HPC} + C_I + C_C - (\sum \alpha_{Hi} C_{Hi}) - (\sum \alpha_{COi} C_{COi})](N_T N_M / HQ_D p) + (MC)_d - N_M (MV)_c - N_T N_M \times [(MV)_h - (MV)_c]\} / [(MC)_d - N_M (MV)_l] \quad (2\text{-}2)$$

Where Q_D is the amount of energy delivered at T_D, p is the simple payback time, H is the hours of operation per year, C_i terms are capital investment terms (HPE: heat pump evaporator; HPC: heat pump condenser; I: interchanger; C: compressor; H: heater; C: cooler), $(MC)_d$ is the marginal cost of the driver utility and N_T and N_M are the thermodynamic and mechanical efficiencies of the heat pump.

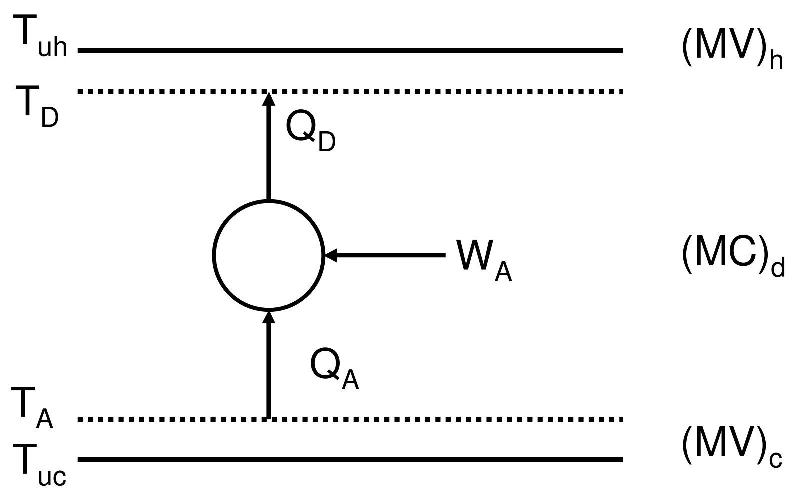

Figure 2-5 Heat pump in a region bounded by two utility levels (Ranade and sullivan, 1988)

For the investment cost calculations, the heat pump is considered as consisting of two heat exchangers and a compressor. The working fluid streams are added to the process stream system, and the investment costs for evaporator and condenser are obtained as a part of the investment cost for additional heat exchanger area. No heat pump model was reported and the COP was calculated by multiplying the ideal COP by a Carnot efficiency. The method was intended to be used both in grass-root designs and in retrofit projects to identify the potential for utility savings and not to identify the actual savings that can be achieved.

Ranade and Nihalani (1986) used the GCC to identify suitable industrial processes for heat pumping and to identify the smallest feasible temperature lift for specified sizes of heat pumps. They showed that appropriate processes have large latent heat loads at nearby temperatures, such as distillation columns and evaporators. The processes were the same as those identified with the unit operation-oriented approach, but with the important difference that the heat pumps were now being correctly integrated thermodynamically. The composite curves were used in order to estimate additional heat exchanger area for calculations of the total investment cost and pay-back period. The presented method was intended for retrofits, but the existing heat exchanger network is not taken into account.

Brown (1987) used the GCC and simplified process flow diagrams to identify the opportunities for heat pumping in the processes and no mathematical optimization methods were used. It was shown that heat pump integration should be considered after utility savings have been economically maximized by process changes and increased heat exchange. A case study of a polyethylene plant was presented. It was observed that the overall best alternative is to integrate the plant with a fractionating plant. The second best choice was a combination of increased heat exchange and heat pump integration, which would save 48% of the utility with a PBP of 13 months. Hot and cold utilities in a reboiler and a condenser were directly replaced by a heat pump, respectively, and thus the HEN was not influenced.

Chappell and Priebe (1989) presented the process integration of different heat pump types with the GCC. The study included classification of heat pumps, economic evaluations, and placement of heat pumps in industrial processes. Pinch analysis was used in the implementation studies. It was shown that even in processes that are highly energy integrated, there is a considerable potential for heat pumping in several industries, but, where maximum energy savings are desired, heat pumping should be considered in the context of overall process integration. The study is done by first increasing the heat exchange and thereafter the uses of heat pumps as well as heat transformers. They used the PBP to compare economically the heat pumps opportunities. No clear methodology/ guidelines were presented to determine the placement, size, and type of heat pumps.

Wallin et al. (1990) presented an optimization methodology for electrically driven heat pumps integration in industrial processes. The method uses the composite curves as a guideline to the correct choice of heat pump. The selection is mainly done by matching the shape of composite curves against the characteristics of several heat pump types. The optimization methodology was presented considering various parameters such as: heat source temperature, heat sink temperature, heat pump capacity and the choice of streams. To show the potential for electrically driven compression heat pumps, two different examples were studied; one with close composite curves which was thought to be a poor heat pump candidate and one with open composite curve which was thought to be a good heat pump candidate. It was found that for both examples, heat pump implementations were advantageous under good economic conditions for the heat pump, i.e. low electricity price, high fuel price and low investment costs. The conclusion that can be drawn is that heat pumps are interesting alternative to decrease the annual costs in the industrial processes, especially when the composite curves are open and operating costs are large. It was shown that for more close composite curves, the possibilities of successfully installation of a heat pump are not that obvious, but the potential should be considered in every case. The most important factors that affect the implementation of heat pumps are the shape of the composite curves, the electricity- to-fuel price ratio, the absolute level of energy prices and the specific heat pump cost. It was shown that the influence of each previously mentioned factor depends on the shape of the composite curves and the combination of the other factors. However, since the method did not consider special constraints of AHPs, it was not appropriate for the implementation of AHPs.

Pendayala and Patwardhan (1991) presented a different method and based it on first selecting the process streams to be used by the heat pump. The aim is to integrate the largest possible heat pump, respecting the pinch rules. The user chooses the streams to be used and the composite curves are used to estimate the maximum allowed evaporator and condenser sizes at the specified temperature levels, which are compared to the sizes obtained from the stream choices. More than one stream on each side of the pinch can be chosen and stream splitting might happen. However, the optimal integration of the heat pump was not discussed and no economic calculations were made.

Linnhoff and Dhole (1992) presented a method for minimizing the shaft work in refrigeration systems based on the exergy composite curves (ECC) and the exergy grand composite curves (EGCC). Those curves are obtained if the temperature axis in the composite curves is recalculated with the Carnot factor, in which the ambient temperature is the high temperature and the refrigeration temperature is the low one. The method can also be applicable for heat pump integration. The procedure is based on the understanding of exergy loss. The area between exergy composite curves and the utility levels are proportional to the exergy losses, which in turn is proportional to the amount of ideal work lost in the heat exchange. The method estimates the shaft work requirement directly from the process stream data without going through the detailed refrigeration design calculations. The shaft work minimization is done by distributing the original and new refrigeration levels so that the area is minimized. In that work, no capital costs are taken into account and new refrigeration levels should be introduced carefully to keep the pay-back period low. The refrigeration machinery is modeled by multiplying the ideal COP by a Carnot efficiency, which is claimed to be sufficient. The method not only offers an estimation technique but a powerful tool for suggesting potential modifications in the system configuration and heat exchanger network.

Wallin and Berntsson (1994) presented a methodology for assessing heat pump opportunities in industrial processes. The methodology takes into account both the industrial process and the heat pump. It was shown that for the correct placement of heat pumps the characteristics of both the industrial processes and the heat pump must be taken into account. For the processes, the pinch temperature, the number of heat exchangers and the shape of the composite curves were found the most important factors. For the heat pump, possible COPs, investment costs and energy prices are the most important factors to be considered. They have developed methods for optimization of the main parameters in a grass-root design and for finding the most appropriate designs in a retrofit situation. It was concluded that there are normally many combinations of size and condensing and evaporation temperatures, which result in the same payback time. It was also shown that, normally as large as possible a heat pump at a specified demand for payback time should be chosen, due to its high annual profit. For a heat pump of a given size, the same payback time was expected to be achieved at both a high temperature lift (low HEN investment costs, high operating costs) and a low one (high HEN investment costs, low annual operating costs) but a somewhat lower annual profit. However, integration of more complex configurations of heat pumps like AHPs and AHTs has not presented.

2.2 Industrial utilization of Heat Pumps

2.2.1 Different Industrial Processes

Several studies have been done to analyze the potential for heat pump integration in different industrial processes. Different types of HPs have been considered such as: compression heat pumps (open, semi-closed and closed), absorption heat pumps (type I and type II). The compression heat pumps have mainly been electrically driven, but some fossil-fuel driven heat pumps are reported. Different industrial sectors have been considered and analyzed. Several of these works are done in cooperation with different companies, and thus information published is small.

According to a study by the International Energy Agency (IEA HPC Brochures, 1993), HPs are used in greater proportion in the forestry sector (68%); next is the food processing industry (24%), and the chemical and petrochemical industries (8%). Textile and metallurgy industries, treatment of wastewater, urban heat, ceramic, and drying biomass together represent about 8% of applications. Most of the time it is Vapour-Recompression Heat Pumps that are the focus of these applications. Absorption HPs occupy an increasing portion of the huge market for coolers used in high capacity air conditioning installations. These machines are also present in certain specialized areas, such as the deep refrigeration of foods.

Estimation of the opportunities for the heat pump integration in evaporation processes of typical industrial processes is presented by Union Carbide Corporation for EPRI (1986). The only heat pump type considered was the open-cycle electrically driven vapour-recompression heat pump. They considered heat pump integration together with other methods for utility savings such as increased heat exchange and evaporator system modifications. They used Grand Composite Curve to analyze the system and to identify heat pump opportunities and the composite curves to calculate the need for additional heat exchanger area for different values of ΔT_{min}. Thirty eight industrial processes in five industrial sectors were considered and ten of these were chosen for further analysis; at least one industrial process from each sector was included. It was observed that in most of the processes, there were different opportunities for both heat pump installations and process modifications. It was shown that when heat pumps are suitable, the utility saving is normally around 20-30% and the pay-back periods for only the heat pump installations varies from below one year to three years. They concluded that even when the process has been heat-integrated, there is a good potential for open-cycle heat pumps (in which the process stream directly enters into the compressor for the desired temperature lift) with acceptable payback times.

The potential for process integration of heat transformers is presented by Berntsson et al. (1989). The study aimed to first identify suitable industrial processes for integration of heat transformers followed by estimation of the potential for utility savings. Twenty nine processes in four industrial sectors were considered. Nine processes were chosen for further analysis, considering that the hot pinch temperatures, between 60 °C and 130 °C, are suitable for heat transformers. The study was done with the aid of the method presented in Wallin et al. (1990). Two sets of costs data were considered and with the favourable costs, the PBPs are below 2.5 years, and with the non-favourable costs the PBPs are below 6.5 years. It was shown that there is a good potential for heat transformer installations in 25% of the originally studied processes.

Schnitzer (1988) studied the potential for implementation of open cycle compression heat pumps and absorption heat pumps in ethanol separation plants. The utility savings were presented up to 54% with process changes, up to 30% with heat pump integration and up to 54% with both techniques used. Although the payback times are not reported, it was claimed that the energy saving scenarios were interesting at the energy price levels of 1988.

A work by TENSA Services done for US DOE was presented by Rossiter and Toy (1988). In that research, heat pump opportunities were studied in a number of typical industries. They made a technical and economic comparison between the process use of closed reverse Rankine cycle heat pumps (which are similar in concept to most domestic heat pumps) and semi-open cycle heat pumps (of which mechanical vapor recompression is an example). Twenty six processes in five industrial sectors were considered, and ten processes were chosen for further analysis. Evaporators and distillation columns are found in several of the chosen processes. The results presented were obtained from the Grand Composite Curve analysis. Results were reported based on the method presented by Ranade and Sullivan (1988). They considered both heat pump integration and increased heat exchange. It was shown that in most processes, increased heat exchange could save 10-40% of the utility demand, and that heat pump integration could be used in a larger number of processes and save comparatively larger amounts of utility. The heat pump installations were analyzed in terms of evaporator size, temperature lift and delivery temperatures. Most evaporator heat loads were between 0.7 and 1.4 MW. Most delivery temperatures were between 93°C and 150°C. Most temperature lifts were between 17°C to 22°C for semi-closed heat pumps and between 28 °C to 39 °C for closed heat pumps. Most heat pump installations had PBPs between 2 and 3 years but no PBPs were presented for increased heat exchange. As conclusion, it was shown that there was a good potential for heat pumping in these industries.

2.2.2 Food Processing Sectors

The food industry has a large variety of processes with operating conditions that are favorable to HP integration. The majority of operations are carried out at temperatures lower than 150 °C. The most common areas of application are pasteurization, sterilization, evaporation, drying, cooking and refrigeration.

According to a study by Lazzarin, installing a VRHP on a malt-drying oven saves 25% of the energy, and yield a four-year return on the investment (Lazzarin, 1995). An absorption HP can also be installed, but the return on the investment is longer. In the beer brewing process, installing a VRHP can save as much as 80% of the energy according to a pilot study (Lazzarin, 1995). The return on the investment is between two and three years. Pasteurizers have a heating and a cooling zone, where a HP can be inserted. To dry fish, it is possible to save in the range of 75% of the energy by installing a HP in the dehumidifier. Drying grain is one of the most energy-consuming food processing operations. It has been demonstrated, however, that the installation of an absorption heat pump can significantly reduce the primary energy demand and produce steam from heat that is useable elsewhere in the process. The feasibility of a multi-purpose heat pump incorporated into an evaporation and crystallization unit in a sugar beet production plant demonstrated that the steam used could be reduced by up to 15% (Scott *et al.*, 1999). Other

VRHP applications in the food industry are reported, for example making buttermilk, sugar starches, chemical products and malt runoff. VRHPs can also be found in the distillation of whisky (Moser and Schnitzer, 1985). There are ways to save energy for slaughterhouses and meat processing plants as well. Energy saving measures that can be put into effect have been evaluated in terms of reducing CO_2 emissions and by considering four different options. It was demonstrated that it is better from an economic point of view to invest in improving the heat exchanger network rather than in HPs, although HPs would reduce the total production of CO_2 by between 5 and 35% (Fritzson and Berntsson, 2006).

In most food processing plants, enormous amounts of thermal discharge from the cooling units could be recovered. The available heat is between 20 and 40 °C and generally cannot be used. If there is a demand for large quantities of hot water, this residual heat recuperated by a HP could be used to preheat fresh water.

2.2.3 Chemical and Petrochemical Sector

In these industries, much of the equipment, such as distillation columns, evaporators, concentrators, dryers and washers lend themselves particularly well to reusing thermal wastes, which would otherwise be lost. The potential installation of an absorption thermotransformer at the fat separation by hydrolysis unit in a fatty acids and glycerol production plant was investigated (Aly *et al.*, 1993). This machine operating with the unusual H_2O/NaOH pair could recover almost half of the thermal content of steam from depressurizing various condensates and reuse it to concentrate or separate unrefined products. The feasibility analysis of using an absorption thermotransformer for the thermal integration of an ethanol distillation unit revealed an interesting possibility for savings (Lazzarin, 1995). It was also demonstrated that the same type of equipment could be used to great advantage to recover latent heat in the steam effluent from drying towers (Currie and Pritchard, 1994).

There are similar opportunities in the petroleum refining industry. One example is the installation of an absorption thermotransformer in a rectification unit to recover the heat of condensation and produce medium pressure steam (Moser and Schnitzer, 1985). Similar uses, but with VRHPs are relatively frequent, for example, in propane and propylene separation units (Moser and Schnitzer, 1985). Absorption coolers are therefore used for cooling and liquefying petroleum gas (Erickson, 1999).

2.2.4 Wood Drying Process

Drying wood improves the wood properties and, thus, its value; the drying temperature and rate are critical factors. Since thermal energy consumption represents between 60 and 70% of the total cost of the wood drying process, energy savings are essential in this industry. Drying by HP is an effective technique to save energy that is frequently used for wood, but also for vegetables and grain (Fessel, 1974; Prasertsan and Saen-saby, 1998; Smadja and Blaise, 1990). The HPs used for this are invariably the conventional VRHPs. The interesting part of this method is eliminating the humidity: the latent heat of water vaporization is recovered by cooling the air in the evaporator of the HP (Figure 2-6). Nevertheless, the economic advantage of this technique is marginal because of the low temperatures for drying, the large investment and the high cost of electricity in some countries.

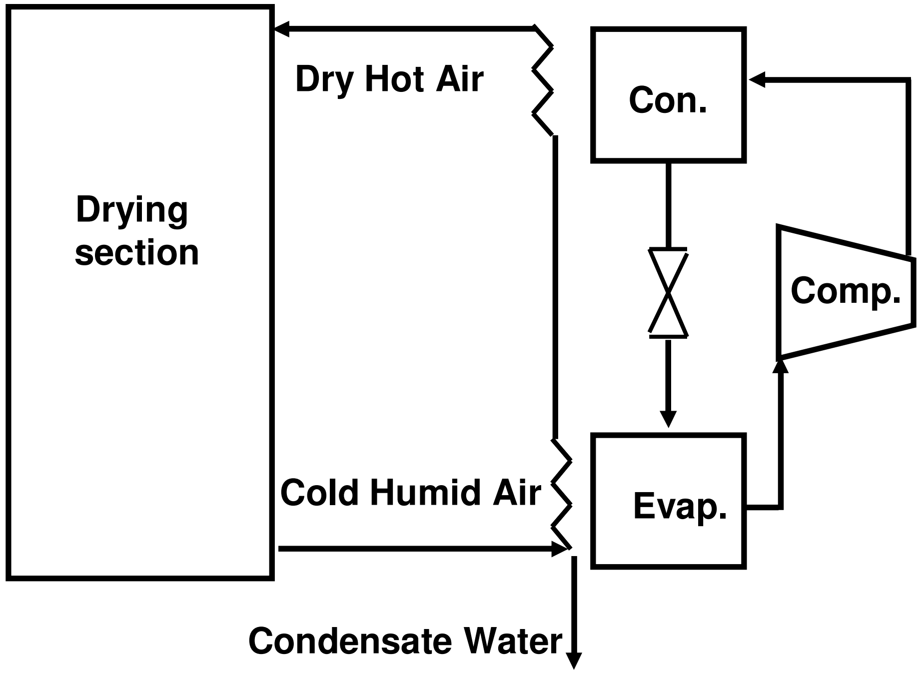

Figure 2-6 Dehumidifying the air to dry wood with an heat pump

Le Lostec et al. (2008) have presented the thermal and economic analysis of a mobile wood chip drying process fitted with an absorption heat pump. Schematic diagram of the drying system with a single-stage absorption heat pump is presented in Figure 2-7. The performance of the dryer coupled with an absorption heat pump was studied in different operating conditions, in order to determine the optimal energy consumption of the dryer. The performance of the system from the energy stand point of the view was compared to the performance of two other systems, direct heating of the air by a wood burning furnace and heating of the air using industrial waste heat. It was shown that single-stage absorption heat pumps can only be implemented when the temperature of the drying air is below 60 °C. When that temperature is higher than 60 °C, the costly double stage absorption heat pump should be implemented. Of the three processes presented, air heating using industrial waste heat was proven to be the most energy efficient option.

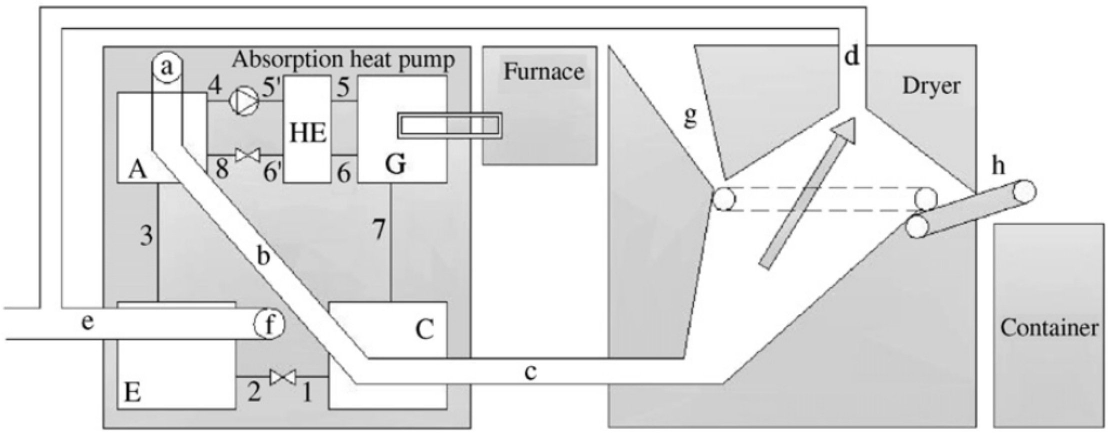

Figure 2-7 The drying system with single-stage absorption heat pump (Lostec et al., 2008)

2.2.5 Pulp and Paper Industry

To the knowledge of the author there are only few reports of actual implementation of AHPs in the P&P industry in the scientific literature. However, the first AHP unit installed in a P&P mill has been in operation for 15 years (Abrahamsson *et al.*, 1995). It is a 200 kW heat transformer that delivers steam at 125 °C using secondary vapour at 85 °C from the liquor evaporation plant. At a pulp factory in Japan, an AHP was installed to recover heat from the alkaline bleaching effluents, at about 50 °C and use it to increase the temperature of water supplied to the boiler from 22 to 80 °C by using 1.1 MW steam (0.4 MPa, 165 °C) (Miura, 1995).

Hester and Stengl (1980) present a conceptual design and economic evaluation of a heat recovery/heat pump system as an integral part of a pulp and paper plant. The basic pulping steps were evaluated to determine the best locations to recover the waste heat and the best locations and best temperatures at which to reintroduce the recovered and enhanced energy into the process. Based upon that analysis, both closed and open heat pump systems were discussed and evaluated to determine working fluids, component sizes, etc. An economic analysis was performed to determine heat pump system capital costs and the amortized cost of the recovered and enhanced energy. The analysis was conducted for a 100-ton/day mill using thermomechanical pulping.

Robb et al. (1983) present a feasibility study for upgrading heat rejected by newsprint processes to displace fossil fuel mainly used to dry the product. They showed that newsprint is dried at lower temperatures than other pulp and paper products so less upgrading is required. The various heat pumps were described and their potential was estimated in terms of fuel displacement and electrical capacity requirements.

Ljung and Scharnell (1985) present how heat pumps can be used for steam or hot water production to improve the energy economy in the pulp and paper industry. It was shown that in a paper mill, where low pressure steam is required for the paper machine dryer, a heat pump can use outlet humid air from the dryer as the high temperature heat source to produce steam. Another good waste heat source in the pulp mill was presented to be the low pressure steam from the last stage of a black liquor evaporator line. It was shown that if only the latent heat in the black liquor steam is recovered, the temperature lift can be kept at a moderate level, and the COP would be rather high at around 3. It was estimated that the steam output in that application would be 15 to 20 MW. It was shown how the economy of a heat pump installation varies with different parameters and that various proposed applications give short pay-back time.

Noel (1985) presents the implementation of cyclotherm[2] heat pump in the pulp and paper industry. He showed that the cyclotherm process produces high pressure steam up to 965 kPa from many low temperature waste heat sources down to 72 °C, and the energy input is a small part of the energy output of the heat pump. The various applications and the economics evaluation was done and it was shown that it has a great potential for energy savings in the P&P industry.

Ljung and Scharnell (1986) present how the humid air from paper machines or low pressure steam from black liquor evaporators can be used as waste heat source for steam production. It was shown how the economy of a heat pump installation varies with different parameters and that various proposed applications give short pay-back time.

Abrahamsson et al. (1994) present two different applications of absorption heat pump in the P&P industry; an evaporation plant in a pulp mill and a paper drying plant in a paper mill. In the first application, optimal energy conservation strategies were investigated using a heat transformer system incorporated with the evaporation plant of the pulping process. Two different working pairs were investigated and the simulation results revealed that H_2O-NaOH was superior to H_2O-LiBr as working pair for that particular application. A process configuration, designed for the largest energy consumer unit in the evaporation plant, with a heat transformer boosting the

[2] Cyclotherm heat pump is a system using heat from dirty steam in an inverse-Carnot cycle. The difference with MVR (Mechanical Vapour Recompression, in which clean steam is produced recovering heat from dirty steam or heat that comes from the refiner) is that an intermediate fluid is used to be compressed and not directly the steam.

temperature of the last vapor stream by 31 °C, would reduce the amount of live steam used in that unit by 18.5%. The pay-off period for that case was calculated as 4.4 years. In a second application, simulation results were presented for different process configurations where an absorption heat pump was suggested to be incorporated in an existing paper drying plant. A total thermal energy amount of 30 MW was needed in the mill, at a temperature of 75 °C, for mixing-pits and district heating. Installing an absorption heat pumping unit, utilizing exhaust humid air at a temperature of 54 °C from three paper machines, would result in a recovery of 12 MW. Three different working pairs were compared and H_2O-NaOH was selected based on the energy index criteria. The pay-off period of that application was calculated at 3.6 years.

Gidner et al. (1995) present the simulation results of an energy efficient evaporation process for the treatment of bleach effluents in the pulp and paper industry. It was shown that due to the very low concentration of the effluent stream, the evaporation process must have a high degree of energy efficiency in order to compete with other treatment alternatives. Optimal energy conservation strategies were investigated where an absorption heat transformer (AHT) unit is integrated with both the existing black liquor evaporation plant and the bleach effluent evaporation process. Different process configurations were simulated using a flow sheeting program, EVAPSYS, developed for simulation of general multiple-effect evaporation processes and absorption heat pump systems. Using real operating data from a major Swedish pulp and paper mill, simulation results were reported for optimum locations of the AHT unit.

Abrahamsson et al. (1997) present optimal energy conservation strategies using different heat pump systems in paper drying. They used real operating data from a major Swedish mill. Simulation results were compared for compressor-driven and absorption heat pump systems. An absorption heat transformer was also investigated. It was shown that a CH_3OH-LiBr double-lift cycle would have a low COP value due to the low temperature of the moist air stream and the restricted temperature of the cooling water available. A total of 30 MW thermal equivalent was needed in the mill at a temperature of 75 °C for mixing-pits, district heating and a log store. Exhaust humid air at a temperature of 54 °C from only three of the paper machines was used in that study. Simulations revealed that installing a mechanical heat pump unit using HFC 134a would result in a recovery of 22 MW due to the temperature level of that application. On the other hand, 12 MW could be recovered with an absorption heat pump. To optimize the operating conditions, H_2O-NaOH was selected as the best of three based on exergy index criteria. Assuming a steam cost of 22 $/MWh and an electricity cost of 32 $/MWh, the pay-off periods was 3.3 and 2.9 years for compressor-driven and absorption heat pump alternatives, respectively.

Costa et al. (2007) studied the energetic and economic incremental impacts of the retrofit implementation of three technological options, cogeneration, absorption heat pump and, trigeneration; using actual energy consumption data and the recorded seasonal consumption fluctuations for a kraft pulp mill. The simple pay back time and the net present value of the options were calculated. The results suggested economic viability for all options investigated. The simple pay back times range from one year for a single heat pump to 2.5 years for a trigeneration system including a heat pump covering 40% of the LP steam demand in the process. Moreover, the net present value was in all cases positive, which indicated that the equipment was viable. The options with high yearly revenues from power sales showed the highest net present values, between 100 M$Can and 120 M$Can. The stand alone AHP achieved the shortest simple pay back time, whereas the trigeneration option reached the highest net present value and therefore the best overall economics.

Costa et al. (2009) presented two preliminary studies of implementation of absorption cycles in a kraft process. Figure 2-8 shows the first case study. It involved integration of a double lift heat transformer into the heat recovery circuit of the wood chips digesters to produce low pressure steam equivalent to 25% of the steam demand of the chemical pulping plant. The AHT provided the 13 t/h of LP steam required by the pulping plant and an additional 4 t/h of steam which was fed to the LP manifold. The investment cost for the installed unit was estimated at 7.6 M$. The minimum price of 12 $/t for steam production leads to a simple pay back time of 4.7 years. With the higher price of 20 $/t the pay back time was reduced to 2.8 years. It was shown that if the savings on fresh water pre-heating were taken into account the pay back time would decrease to 1.6 years. The second studied case concerned the installation of a double lift chiller to replace the barometric condenser in the bleaching chemical making plant (Figure 2-9). Because of the high cycle efficiency of the chiller, the need for cooling water would decrease from 845 t/h to 757 t/h, thus saving 10% of cooling tower capacity. The investment for the installed unit amounted to 236 k$. The savings of 1.65 t/h of MP steam at 12 $/t would lead to a simple pay back time of 2.6 years. At 20 $/t the pay back time would estimated as 1.6 years. Both proposals would save substantial MP steam, but in order to comply with the Pinch Analysis thermal requirements, alternate heat sources or sinks would have to be used.

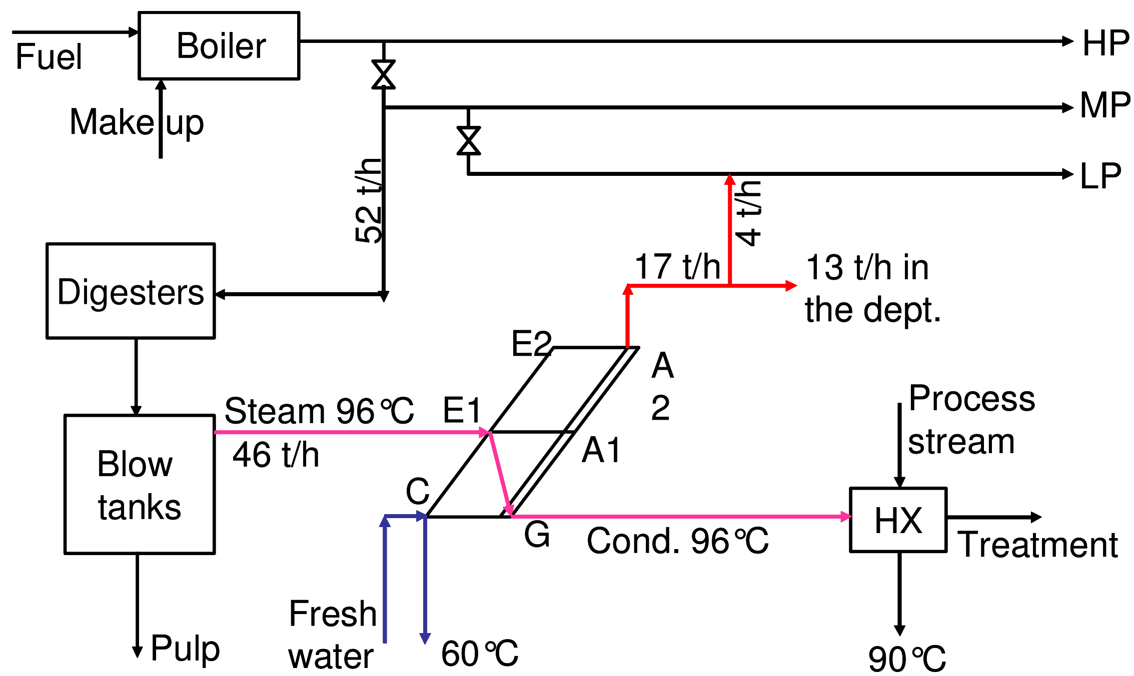

Figure 2-8 Installation of a double lift AHT in the blow tank heat recovery circuit (Costa et al., 2009)

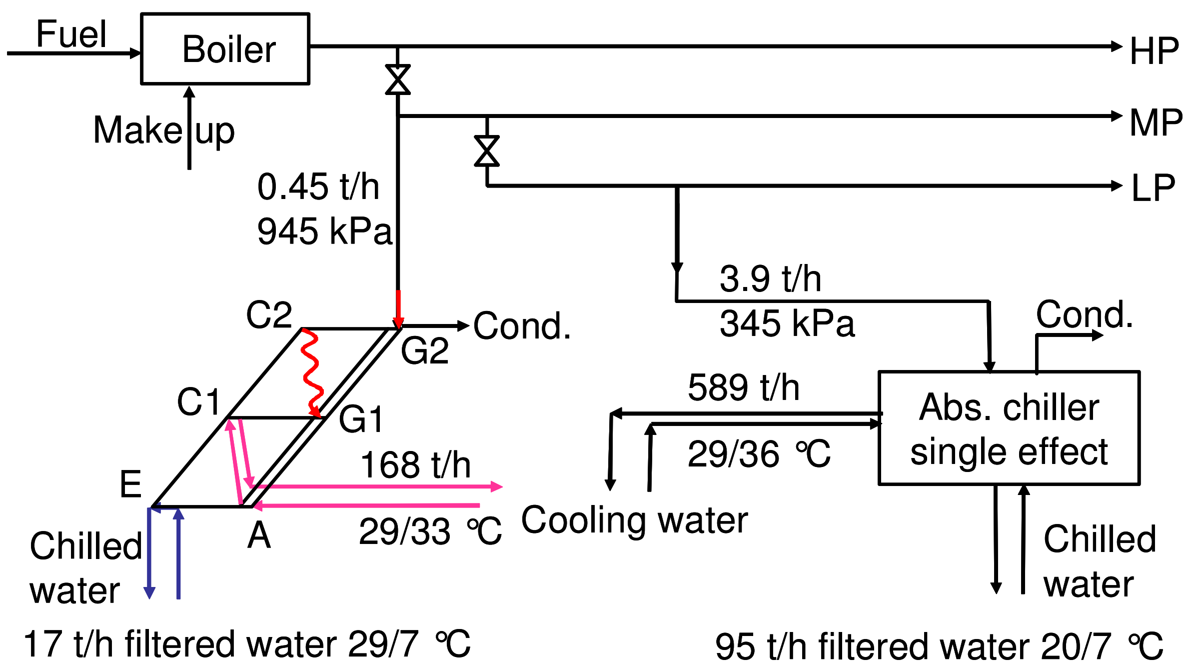

Figure 2-9 Implementation of a double effect AHP to produce chilled water (Costa et al., 2009)

2.3 Simulation models and experimental evaluation of AHPs

In this section different available models and experimental evaluation of AHPs are introduced. Most of the current residential and commercial absorption systems are based on the single-effect cycles. Many studies have been performed in the last three decades to analyze the performance under different operating conditions and employing advanced cycles. Eisa et al (1986) classified more than 400 references of AHP systems on the basis of cooling and heating, and on the basis of the different working fluid pairs.

Grossman and Zaltash (2001) presented a computer code, ABSIM (ABsorption SIMulation), for simulation of absorption systems in a flexible and modular form. Development of the ABSIM program was started in the 1980s. The model can be used to investigate various absorption cycle configurations with different working fluid pairs. The US Department of Energy (DOE) had supported that research under an Advanced Cycles/Advanced Fluids program, expected to evaluate the promising ideas for new cycles and working fluids. Figure 2-10 shows a schematic description of the ABSIM structure. The program calculates the thermodynamic property at each state point and the heat duty of each component based on the user’s defined operating conditions, configuration and working fluid pair. ABSIM is based on unit subroutines containing the governing equations for the system's components and on property subroutines containing thermodynamic properties of the working fluids.

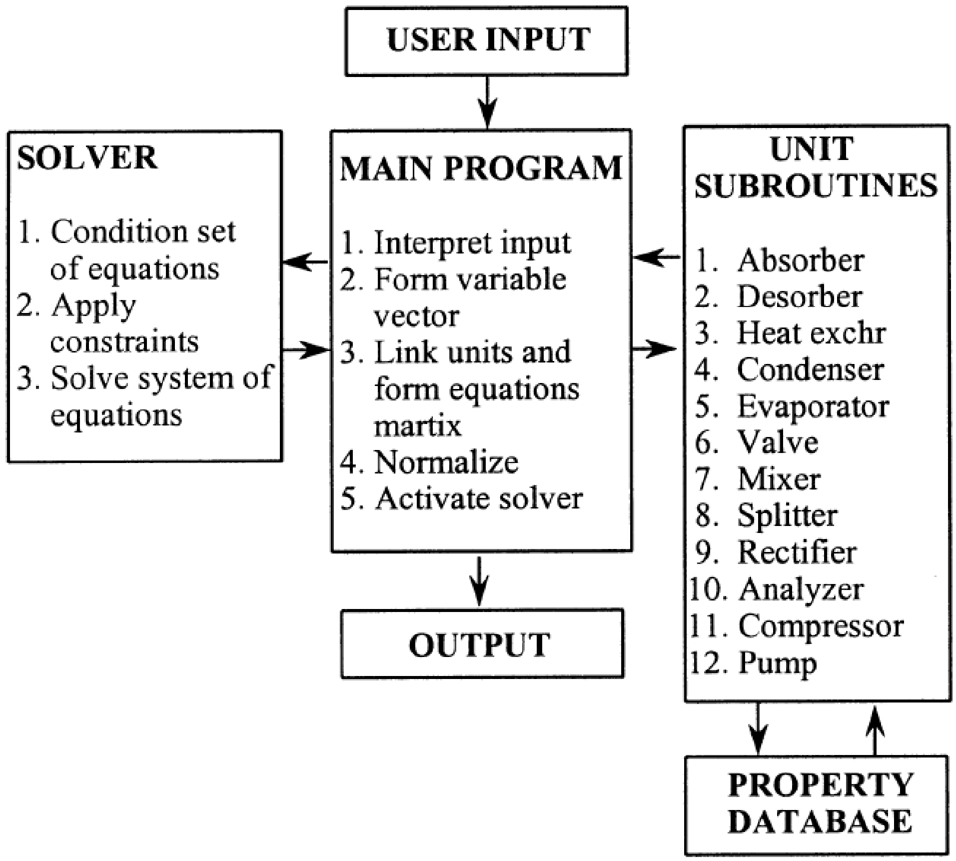

Figure 2-10 Structure of the ABSIM computer code (Grossman and Zaltash, 2001)

Gordon and Ng (1995) presented a general thermodynamic model for cooling devices and applied that to absorption chillers. It was shown how the chiller coefficient of performance should depend on cooling rate and key system variables. They presented an approximate formula for COP, equation 2-3.

$$1/COP=[\frac{T_{Cond}^{in}-T_{evap}^{out}}{T_{evap}^{out}}][\frac{T_{gen}^{in}}{T_{gen}^{in}-T_{cond}^{in}}]+[1/Q_{evap}][\frac{T_{gen}^{in}}{T_{gen}^{in}-T_{cond}^{in}}][A_1-A_2\frac{T_{cond}^{in}}{T_{gen}^{in}}] \quad (2\text{-}2)$$

where the constants A_1 and A_2 characterize the irreversibilities (entropy generation) of a particular chiller. They have compared model predictions against performance data from literature and their own experimental measurements. Their results agreed well with experimental and manufacturer catalogue data.

Ng et al. (1999) presented a simple thermodynamic model for an absorption chiller. The performance of the chiller, as described by 1/COP, is expressed in terms of dominant external and internal losses and internal entropy generation in the components. They presented evidence from experimental facility to show that the absorption behavior is reached by using of three key competing losses: the finite-rate heat transfer losses, the internal dissipative losses and heat leaks. Through the measured data of a real chiller, it was shown that the presence of external and internal dissipative losses (entropy generation) are the key to present the correct trend of commercial absorption heat pumps. The effect of heat leaks was found to be generally small except at low cooling loads. The model was claimed to be a useful tool for thermodynamic modeling, in particular, for designing absorption chillers.

Romero et al. (2001) compared the theoretical performance of the modeling of an absorption heat pump operating with $LiBr/H_2O$ and an alternative aqueous ternary hydroxide mixture. The aqueous ternary hydroxide working fluid consists of potassium, sodium and caesium hydroxides in the proportions 36:40:24 (KOH:NaOH:CsOH). They presented plots of Carnot coefficients of performance against the important temperatures of the system (generator, absorber, condenser and evaporator temperatures). Similar coefficients of performance were obtained for both mixtures; however, it was shown that the system operating with the alternative mixture can operate with a higher range of condenser and absorber temperatures and the heat delivered by these components can be easily removed by air. It was also found that the system with hydroxide blend may operate at higher generator temperatures in comparison with lithium bromide system.

Lee and Sherif (2001) presented thermodynamic analysis of a $LiBr/H_2O$ absorption system for both cooling and heating applications. Simulation results on ABSIM were employed to determine the COP and the exergetic efficiency of the absorption system under different operating conditions such as the heat source, chilled water and cooling water temperature. For cooling applications, it was found that a low cooling water temperature yields both a higher cooling COP and higher exergetic efficiency. It was presented that the AHP operating with the higher chilled water temperature has a better cooling COP but smaller exergetic efficiency. It was shown that, increasing the heat source temperature can improve the cooling COP, but as the heat source temperature increase beyond a certain threshold, the COP decreases. For heating application, it was evident that the heat source temperature will increase both the heating COP and the exergetic efficiency.

Joudi and Lafffta (2001) developed a steady state simulation model to predict the performance of a $LiBr/H_2O$ AHP. Their model is based on mass and energy balances and heat and mass transfer relationships for the cycle components. A computer program was developed and the effect of various operating conditions on the machine performance was studied. A new model for representing of absorber was introduced considering simultaneous heat and mass transfer. The performance of the components; generator, absorber, evaporator and condenser were simulated independently. The overall system performance was then evaluated by combining the component models. All components of the system were considered to be shell and tube exchangers with counter current streams. No validation with experimental results was presented and the simulation results were compared qualitatively with other works from literatures and good general agreement was obtained.

Goodheart et al. (2002) presented the performance and economics of large (600ton, 2110kW) single-effect and half-effect absorption chiller systems as a function of hot water temperatures. A simulation model was developed and was calibrated with data from a chiller manufacturer. In the model, the performance of the system was described with the mass and energy balances and heat transfer equations for the internal and external streams. The entire set of equations was solved with a commercial equation solving package, called EES, which includes thermodynamic properties for water and LiBr-H_2O solution. It was shown that, as the temperature of the hot water entering the generator was lowered from the design value of 110 °C, the single-effect system operates at nearly constant COP of approximately 0.65 and capacity decrease precipitously. At the same conditions, the half effect cycle can maintain capacity, but it would have lower COP, which means more energy is required in the generator and more energy is rejected through the cooling tower. It was concluded that from the economic point of view, the half effect system is more favorable than the single effect system for inlet hot water temperatures lower than 93 °C. The half effect system was not found economically competitive with electrically driven chillers on a five-year life cycle basis, even if the hot water is available free of cost. The cooling tower was found to be a major cost for both systems which was estimated 50% of the total cost for all the chillers investigated.

Florides et al. (2003) presented a method to evaluate the characteristics and performance of a single stage LiBr/H_2O AHP. They also presented the necessary heat and mass transfer equations and appropriate equations describing the properties of the working fluid pair. They developed a computer model and performed a sensitivity analysis. It was assumed that the heat to be provided for the generator can be based on the heat of vaporization of pure water, increased by about 23%

in a typical design; no major assumptions was made for the other system components. They showed that the greater the difference between the absorber LiBr inlet and outlet percentage ratio is, the smaller would be the mass circulation in the absorber. They presented information on designing the heat exchangers of the LiBr/H_2O AHP. The calculated theoretical values were compared to experimental results for a 1 kW nominal capacity unit. The cost for a 10 kW AHP was estimated to be C£ (Cyprus Pounds) 4300. The total cost of an AHP together with all necessary devices and installation cost was estimated as C£ 4800.

Asdrubali and Grignaffini (2005) presented an experimental evaluation of the performance of a LiBr/H_2O AHP under different service conditions. The Japanese-manufactured (Yazaki) experimental single stage absorption heat pump was used for the experimental measurements. The presented machine was water cooled and it was supplied by hot water produced in an electrical boiler. The machine performance was tested by varying the flow rate and the temperature of the supplying hot water. They presented a simulation code developed for their particular plant as presented in Figure 2-11. They compared their experimental data with simulation results and a good agreement was presented. The results showed that the absorption machine can work, with acceptable performance, with low input temperatures of about 65-70 °C.

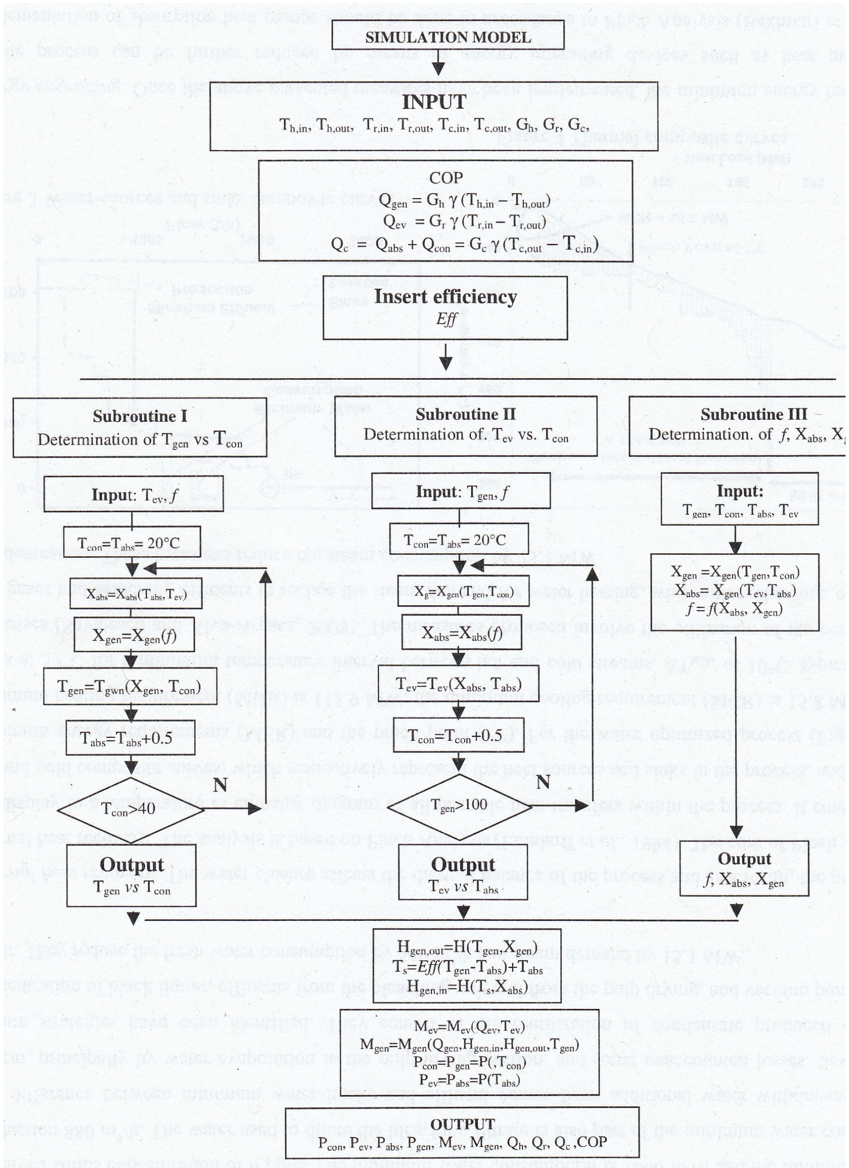

Figure 2-11 Block diagram of the simulation code for the absorption heat pump (Asdrubali and Grignaffini, 2005)

In order to characterize the performance of the AHP, a 33 kW prototype of a single effect $LiBr/H_2O$ AHP was built through an international collaboration (Jahnke, 2005; Jahnke *et al.*, 2005; Neuhann, 2003). To determine limits of operation of this prototype and to characterize the performance of the machine, Jahnke et al. (2005) performed 16 selected experiments based on a partial fractional design with seven variables; those variables were variables were the temperature and flow rate of chilled, cooling and hot water as well as the internal solution mass flow rate. The respective effects of the independent variables on the COP and the heat fluxes in the main four heat exchangers were computed. The variations of COP were very large (from 0.1 to 0.75); while some results were close to the design COP of 0.75, others were out of bound. The experimental results were analyzed using a statistical software. The temperature of chilled, cooling and hot water and, the flow rate of cooling water and hot water were found to be the most influential operating parameters. A mathematical model of the COP and the heat fluxes as a function of the input variables has been determined. The first-order obtained model by regression analysis for the COP is given by:

$$\begin{aligned} COP &= 0.4093 + 0.0124 * T_{Evap} - 0.0328 * T_{Cool} \\ &+ 0.0045 * T_{Gen} + 0.0091 * \dot{m}_{Cool} + 0.0027 * \dot{m}_{Gen} \\ &+ 0.0008 * \dot{m}_{Evap} + 0.014 * \dot{m}_{Sol} + \varepsilon \end{aligned} \tag{2-4}$$

The same procedure was applied to calculate the equation parameters for the heat fluxes. Table 2-1 gives the individual values of the coefficients for each response variable.

Table 2-1 First-order coefficents of the 7 input variables (Jahnke et al., 2005)

	a0	a1	a2	a3	a4	a5	a6	a7
COP	0.4093	0.0124	-0.0328	0.0045	0.0091	0.0027	0.0008	0.0143
Q_{Evap}	-6.9788	0.2187	-0.4589	0.1226	0.2334	0.0992	-0.0249	0.7146
Q_{Abs}	-10.9376	0.1511	-0.4323	0.1683	0.2397	0.1386	-0.0233	1.0660
Q_{Cond}	-8.1491	0.2116	-0.4042	0.1605	0.2396	0.1595	-0.0501	0.2382
Q_{Gen}	-14.4082	0.1513	-0.3479	0.2180	0.2602	0.1999	-0.0426	0.5821

2.4 Critical Review

Based on the presented literature, there is a considerable amount of knowledge available on how to use traditional compression based heat pumps in process where the knowledge about the process integration and Pinch Analysis is utilized and the issue of appropriate positioning of heat pumps is well established. As stated here, less work is done on absorption heat pumps and their integration in industrial processes has not yet been fully exploited due to the lack of clear implementation procedure for this technology; thus, there is an incentive for the development of a systematic procedure to cover this area.

As stated, the P&P is Canada's premier industry characterized by very large energy requirements and competition in this industry has become fiercer over the past few years; high energy consumption has therefore become an issue. Also, Canada is one of the countries that signed the Kyoto protocol with the aim of reducing green house gas emissions by 6% below the value of 1990. For these reasons, the P&P industry must make more efforts to improve energy efficiency and, subsequently, reduce its thermal energy consumption and gas emissions (GHG). It seems that implementation of absorption heat pumps is an appropriate way to reach these aims. As presented before, the previous attempts to implement AHPs in the P&P industry did not consider the Pinch Analysis and were not based on a systematic procedure.

Single-stage AHPs have been investigated in the past and some analytical models have been developed for different purposes. However, most of the developed models are restricted to a specific system and are empirically derived. In some other cases, they are overly simplified; thus there is an incentive for the development of a simple and accurate simulation and dimensioning model to cover this area.

2.4.1 Specific Objectives

Based on the presented literature review, this research has the following approach as shown in Figure 2-12. First, a systematic methodology for the integration of AHPs in a process is developed. A design and dimensioning model is also developed as a part of the whole methodology. Finally, the developed method is used to identify general opportunities for the optimal implementation of AHPs in the Kraft process.

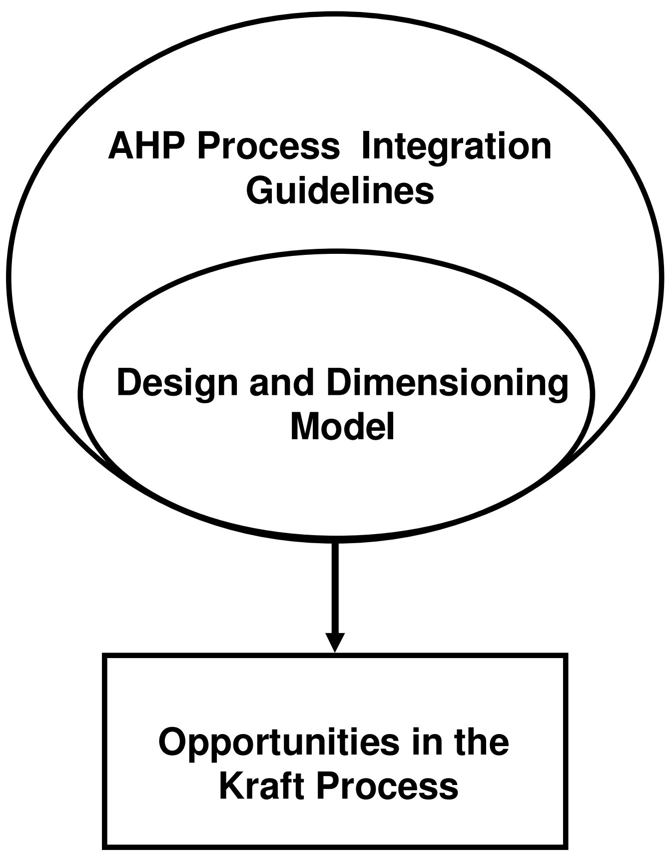

Figure 2-12 Overall methodology approach

This research has the following specific objectives:

- To develop a methodology of implementation of absorption heat pumps in a process based on the results of Pinch Analysis.

- To develop and validate a simple and reliable dimensioning and simulation model for AHPs on the basis of the literatures and experimental data.

- To identify the opportunities and different possibilities of placing absorption heat pumps in a Kraft mill and detail design and economical analysis of them.

CHAPTER 3 - OVERAL METHODOLOGY APPROACH

As introduced in section 2.4.1 (Figure 2-12), the overall methodology consists of three stages. The outer ring represents the AHP process integration guidelines. In this step, a systematic methodology for the integration of AHPs in a process is developed. The method considers: the process, the thermodynamic properties of AHPs, the design aspects and the practical constraints. The inner ring represents the development of a design and dimensioning model of H_2O-LiBr absorption heat pumps. It will be used as the last step of the developed methodology. In order to characterize the performance of the AHP and give an experimental dimension to the project, an experimental analysis of a laboratory single-stage H_2O-LiBr is performed and the steady-state simulation results of the model are compared with experimental measurements. Finally, the capability of the methodology is illustrated by identifying general opportunities for the optimal implementation of AHPs in the Kraft process. It shows that even for a fully energy and water optimized mill, there is still a potential for further utility savings.

An available simulation on CADSIM Plus has been used for data extraction (Mateos, 2009). It is a software specialized in P&P processes, broadly used in Canada.

ASPEN HXNET was used to draw the composite curves and for computation of the HEN and HEN reconfiguration.

MATLAB has been used for the AHPs simulation and dimensioning model.

Chapters 4 to 6 present the main scientific findings of this work and represent the core of the thesis. Each of these chapters consists of an article that has been submitted to a peer-reviewed journal. The following is a brief description of each chapter:

- Chapter 4 provides a systematic methodology for the integration of AHPs in a process. To accomplish this, Guidelines are formulated for the proper selection of heat sources and sinks that will maximize the benefit derived from heat pumping, while respecting process constraints and operating requirements of the AHP. The methodology relies on data extracted from a Pinch Analysis of the plant. The advantages and outputs of the methodology are exemplified using an AHP implementation in a Kraft pulping process.

- In Chapter 5, implementation of AHPs in a Kraft pulping process is studied using the developed methodology. Two generic opportunities are identified for an energy and water optimized mill: (i) integration of a double lift chiller in the bleaching chemical making plant to produce chilled and hot water simultaneously, using MP steam as the driving energy and, (ii) installation of a single stage heat pump to concentrate the black liquor and produce useful hot water by upgrading heat from the bleaching effluent and using MP steam as driving energy. The simple payback time and net present value are calculated to evaluate the interest of such implementations.

- Finally Chapter 6 presents the experimental and simulation analysis of a laboratory single-stage H_2O-LiBr absorption heat pump. The machine performances, as described by the coefficient of performance and cooling capacity, are measured at different flow rates and temperatures of the external cool and hot water loops and for different temperatures of produced chilled water. A simulation model of the steady-state operation of the heat pump is developed and experimental measurements of its performance are compared with simulation results. The capability of the model is also illustrated by dimensioning an absorption heat pump implemented in a Kraft process.

CHAPTER 4 - RETROFIT OF ABSORPTION HEAT PUMPS INTO MANUFACTURING PROCESSES: IMPLEMENTATION GUIDELINES

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Keywords: Absorption heat pump, process integration, energy upgrading, pulp and paper.

4.1 Presentation of the Article

In this article, a systematic method for the retrofit implementation of absorption heat pumps into manufacturing processes as the first specific objective of this research is presented. This article is submitted to the Canadian Journal of Chemical Engineering.

4.2 Abstract

Integration of absorption heat pumps (AHP) in industrial processes has not yet been fully exploited due to the lack of clear implementation guidelines for this technology. In this work, a systematic methodology for the integration of AHPs in a process has been developed and is presented. Guidelines are formulated for the proper selection of heat sources and sinks that will maximize the benefit derived from heat pumping while respecting process constraints and operating requirements of the AHP. The principles of AHP operation and its efficient process integration are thus described. The methodology relies on data extracted from a Pinch Analysis of

the plant. The advantages and outputs of the methodology are illustrated using an AHP implementation in a Kraft pulping process. Two realistic implementation options are presented along with their detailed design and preliminary economic evaluation.

4.3 Introduction

Heat pumps (HP) are thermal machines used to increase the temperature at which a certain amount of heat is available. Absorption heat pumps (AHP) are emerging as a potential alternative to the more common vapour recompression heat pumps (VRHP). AHPs use the effect of pressure on the absorption-desorption cycle of a solution to create a temperature increase of the available heat, that is, the temperature lift. They are thermally driven and, when judiciously positioned into an industrial process, they can be operated in such a way that heat is pumped and energy is recovered at the same time.

Early methods to identify and evaluate heat pumping opportunities generally used a unit operation-oriented approach. Large utility consumers, such as evaporators and distillation columns, were easily identified and a heat pump could be integrated within the chosen unit (Eisa et al., 1987; Ferre et al., 1985; Flores et al., 1984; Supranto et al., 1986). More recent work introduced a complete process approach for heat pump integration and showed the importance of taking into account the thermodynamics of process integration, particularly Pinch Analysis (Linnhoff, 1993). The integration procedure for some types of heat pumps, such as conventional VRHP, is well defined and discussed in the literature (Chappell and Priebe, 1989; Ranade, 1988; Ranade *et al.* 1986; Wallin *et al.*, 1990; Wallin and Berntsson, 1994). It is shown that for a correct assessment of heat pumping opportunities, a full understanding of the thermodynamics and economics of both processes and utility systems, together with the associated interactions is needed. However, integration of more complex configurations like absorption heat pumps and absorption heat transformers in a process has not been thoroughly investigated. Marinova et al. (Marinova *et al.*, 2007) presented guidelines for the implementation of a trigeneration unit (cold, heat, and power production simultaneously) in a Kraft process. They clearly showed that both process and AHP constraints must be considered simultaneously to ensure proper implementation and energy gains.

In view of many failed or deceiving AHP process integrations, yielding no or very little benefit, guidelines are formulated for the most efficient retrofit integration of an AHP into a process, simultaneously considering thermal and thermodynamic constraints from both the process and the AHP systems. The integration should begin by extracting the process stream data and ideally proceed by considering the heat exchanger network (HEN) to maximize the internal heat recovery, which generally is a more attractive proposition than the implementation of heat pumps (Sama, 1996; Yee and Grossman, 1990). This preliminary phase, while strongly suggested, is however not mandatory for the correct implementation and application of the method described here.

4.4 Working Principle of Absorption Heat Pumps

Numerous technical and scientific publications have been dedicated to the fundamental principles, operation and design of heat pumps (Costa *et al.;* Herold *et al.*, 1996; Ziegler and Riesch, 1993). In order to produce the temperature lift in a HP, a refrigerant fluid is circulated between the evaporator (E) and the condenser (C) which operates at a higher temperature and pressure (Figure 1a). The available heat, which is generally not purchased, is fed into the heat pump through the evaporator and released at the condenser. The circulation of refrigerant between the evaporator and the condenser is obtained by means of a compression device, a compressor in the case of VRHP (Figure 1a); this type of heat pump is driven by high quality energy (shaft work or electricity). The compression system of an AHP consists of a secondary loop (Figure 1b) in which a binary solution is circulated between the absorber (A) operating at the same pressure as the evaporator, and a desorber, generally called the generator (G). The refrigerant is the most volatile component of the binary solution. The vaporized refrigerant exiting the evaporator is absorbed into the solvent-rich solution in the absorber, causing the release of additional useful heat, Q_A. The weak solution thus formed is pumped into the generator where the refrigerant is desorbed at a higher pressure and temperature under the effect of the driving force, supplied as heat, Q_G. In a real machine, a fifth heat exchanger (solution heat exchanger; SHX) is inserted between the generator and the absorber to preheat the weak solution with the rich solution, thus reducing the demand for high quality driving energy. This configuration of an AHP is also referred to as an AHP type I. When an AHP is properly integrated into an industrial process, the driving energy is generally not purchased, and represents an additional economic incentive.

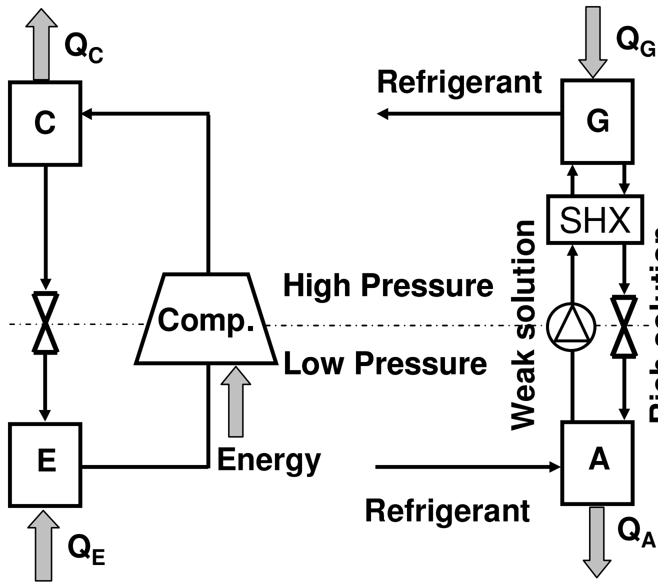


Figure 4-1 Principle of heat pumping: a) VRHP; b) AHP

Some of the working fluid pairs that have been proposed for industrial uses of AHPs include H_2O/LiBr, H_2O/H_2SO_4, NH_3/H_2O, H_2O/NaOH, H_2O/glycerol, H_2O/nitrate salts and H_2O/zeolites. The most commonly used working pairs for industrial applications are NH_3/H_2O and H_2O/LiBr with NH_3 and H_2O as the refrigerants respectively.

If the high and low pressure zones of the machine are reversed, the driving heat is supplied at the evaporator-generator pair, at the intermediate temperature level, while the useful heat is released by the absorber at higher temperature and by the condenser at lower temperature; this configuration is the absorption heat transformer (AHT) or absorption heat pump type II (Figure 2). For the thermal analysis of an AHP, it is convenient to represent the two circulation loops in the vapour-liquid phase diagram of the binary solution (Figure 3a), which shows the thermal operating constraints imposed on the system, the pure refrigerant evaporation line on the low temperature side and the solvent crystallization line on the high temperature side. This type of diagram led to a very simplified schematic representation of AHPs which illustrates how the machine operates (Figure 3b).

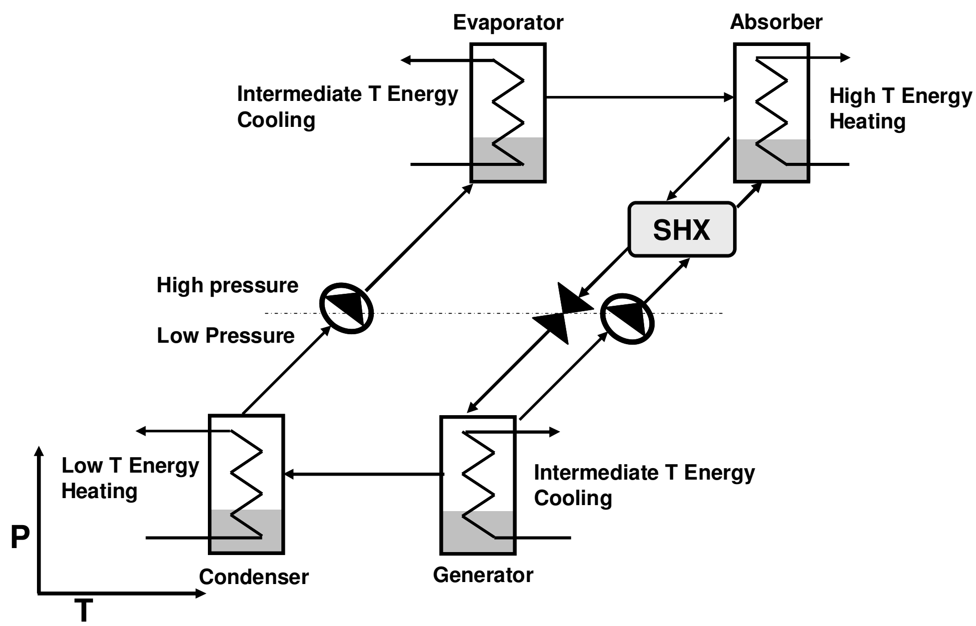
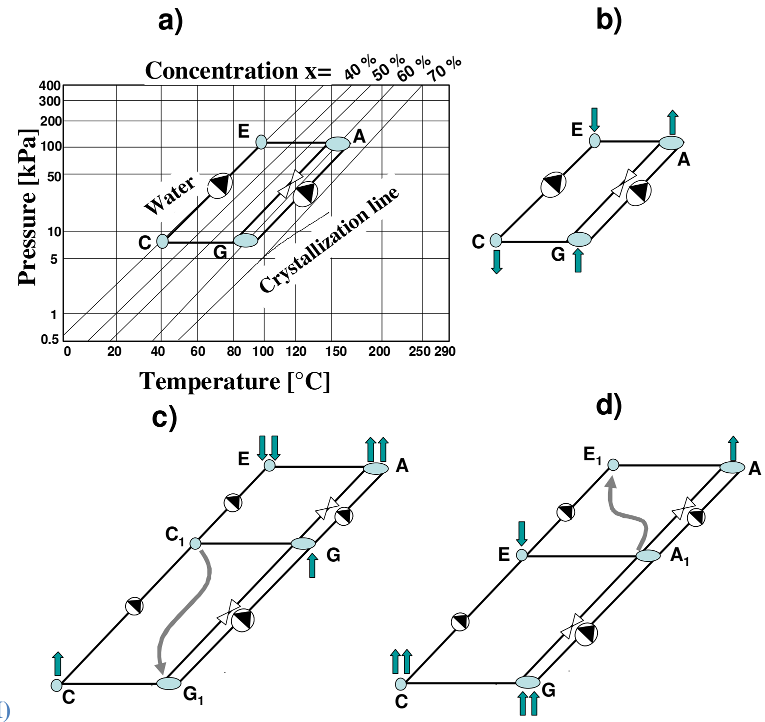


Figure 4-2 Schematic of an AHT (AHP type II)



II)

Figure 4-3 AHT representations; a) In H_2O/LiBr phase diagram, b) Schematic of AHT c) Double effect AHT, d) Double lift AHT

To overcome the limitations imposed by the thermodynamics and the irreversible effects associated with heat transfer, the coupling of several cycles with internal heat transfer is also possible. To achieve higher cycle performance, a double cycle that takes advantage of the higher availability (exergy) associated with a higher temperature input is used. The double effect cycle represents such a cycle variation (Figure 3c). Other high-performance cycles are also possible and are presented in the literature (Ziegler, 1999; Ziegler and Riesch, 1993). It is possible to reach higher temperature lifts by coupling cycles, especially for low temperature applications; the double lift cycle represents one such cycle variation (Figure 3d).

The coefficient of performance (COP) is used to define the heating performance of heat pumps. It is defined as the ratio of the heat delivered by the heat pump to the driving energy; for an AHP type I, COP = $(Q_C + Q_A) / (Q_G)$. Typical COP values for different configurations are presented in (Ziegler and Alefeld, 1987).

4.5 Heat Pump Integration in a Process and Pinch Analysis

Pinch Analysis is a technique used to maximize internal heat recovery within a process. Its scientific principles and utilisation procedures have been presented in engineering manuals (Linnhoff, 1993; Linnhoff et al., 1994). Results are often displayed as composite curves. The development of the composite curves, their relation in the T vs. ΔH graph and their significance have been described elsewhere (Kemp, 2007). The thermodynamics of Pinch Analysis dictates a fundamental rule: there must not be transfer of heat from above to below the pinch point. If this happens, the process suffers a double penalty: the simultaneous increase of the cooling and heating requirements of the process. On the other hand, an HP must transfer heat in the opposite direction from below to above the pinch point; so it should be integrated in such a way that the heat source is situated where there is an excess of heat, i.e. below the pinch, and the heat sink where there is a need for heat, i.e. above the pinch (Bakhtiari *et al.*, 2007; Ranade and Nihalani, 1986).

The composite curve diagrams can be used to position heat pumps in the process so as to maximize the overall energy benefit. The composite curves diagram are preferred rather than the grand composite curve diagram (Kemp, 2007), because it represent all streams being involved in the process and thus, later can be used to redesign the HEN. Figure 4a illustrates the simple case of a VRHP. When, as shown, the heat is supplied to the evaporator by a hot stream below the pinch point, the minimum cooling requirement (MCR) is reduced by the amount Q_E; similarly, if the heat is released by the condenser to a cold stream above the pinch point, the minimum heating requirement (MHR) is reduced by the same amount Q_C. It can easily be verified that when this condition is not met, the benefit of implementing the heat pump is practically nil. The case of an AHP is slightly more complex, because there can be three points of heat exchange between the process and the HP (Figure 4b). The condenser and absorber must release their heat above the pinch point to reduce the MHR; the generator which is at a higher temperature can thus only be above the pinch point and, to reduce the MCR, the evaporator must be below the pinch point as in the case of the VRHP. The overall energy gain is effectively the same as for a VRHP of equal power, but the need for shaft work is eliminated. The correct positioning of an AHP type II (AHT) is also presented in Figure 4c.

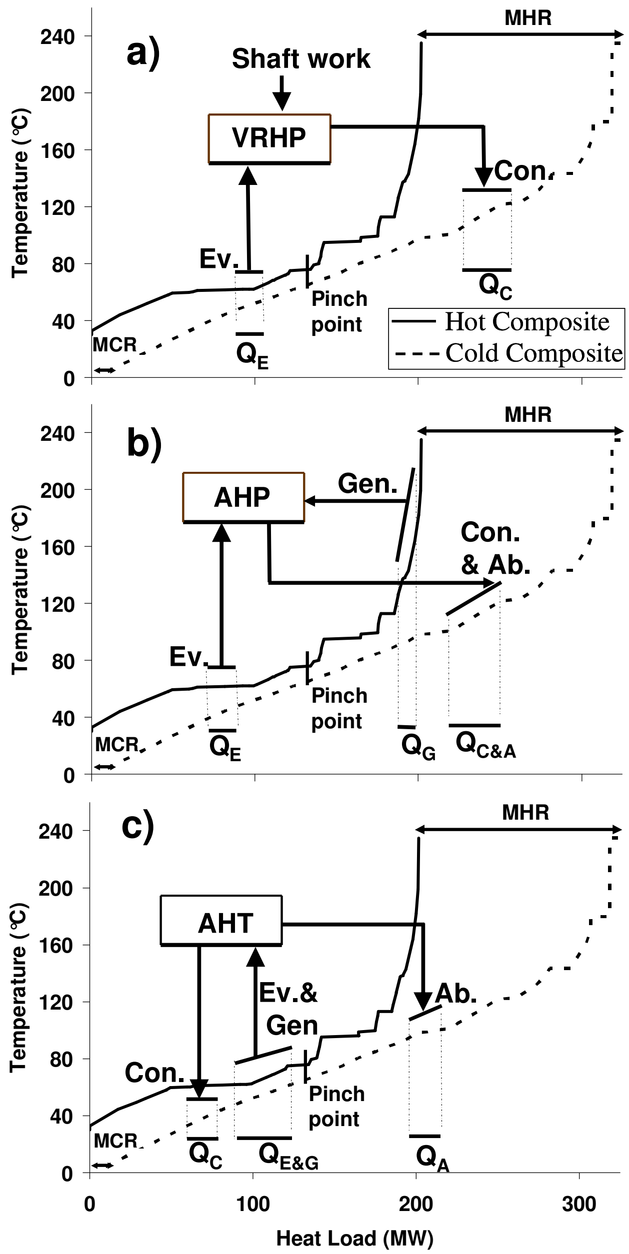


Figure 4-4 Correct positioning of HP based on Pinch Analysis; a) VRHP b) AHP type I c) AHP type II (AHT)

The difficulty when implementing an AHP in a process is to determine the streams that will best fit the AHP, and what is the best configuration and working pair, taking into account the working fluid phase diagrams and other design and technical constraints. The objective of the proposed method is to answer those questions and to show how the process, the working fluid pair and the types of AHP are intimately linked.

4.6 Methodology for AHP Positioning in a Process

The method considers both the process and AHP thermodynamics and design aspects. In this section, the process aspects to be considered for the implementation of AHPs are presented followed by AHP design aspects and interrelation between both. A list of tasks is introduced for the optimal integration of an AHP in a process.

4.6.1 Process Analysis

The method assumes that a Pinch Analysis of the process has been performed and thermal data on all streams are available. Since heat pumping is more expensive than heat exchanging, the HEN should first be considered and the process should be heat-integrated. It should be mentioned that other energy recovery methods such as condensate return and water closure are recommended before pinch analysis and heat pump implementation. In general, they are cheaper energy recovery measures and they help reduce the pinch point temperature, which facilitates the implementation of AHPs.

The process streams that could be used by the heat pumps should be selected on the basis of their location in the composite curve and their heat loads. In the first attempt for the implementation of an AHP, streams which are heated or cooled by a utility are selected, because their heat content is wasted to the environment and not used in the internal heat recovery network. HEN reconfiguration and the utilization of aftermath streams may be considered in later attempts as will be discussed later. The available streams can be classified into 4 groups; heat sources (H) and sinks (C) (hot and cold streams) above (A) and below (B) the pinch point, i.e., HA, HB, CA, CB (step 1). As shown in Figure 4, all available streams above or below the pinch point have the potential to be used in the AHPs. Some HB streams could be used as heat sources for an AHP type I and CA streams as heat sinks for an AHP type II. Some HA streams could be used as the driving energy for AHP type I and CB streams as low temperature heat sinks for an AHP type II.

Since the objective of a heat pump is to reduce MHR and MCR by energy transfer from below to above the pinch point, each potential streams combination for an AHP contains a CA and a HB stream. An opportunity stream (a CA or a HB stream) should be selected to produce a catalogue of different stream combinations between the opportunity stream and the corresponding available CA or HB streams. The opportunity stream may be chosen in order to solve a particular heating or cooling requirement identified in a process. Streams with temperature closer to the pinch point and high heat loads are most desirable whenever feasible, because they require a lower temperature lift while supplying a large amount of energy to be upgraded. For each stream combination, the maximum evaporator temperature is estimated taking into account the target temperature of the HB stream and the heat exchanger temperature approach ΔT. The minimum condenser/absorber temperature (for AHP type I) or absorber temperature (for AHP type II) is estimated for each of selected heat sink stream and chosen ΔT as will be discussed in the Interactions Analysis section. In order to produce a set of thermodynamically feasible options, some of the choices should be eliminated because of process technical constrains such as physical distances between streams, and maximum or minimum allowable temperature and pressure of process streams.

4.6.2 AHP Analysis

In this methodology, the phase diagrams of different working pairs are used to estimate generators temperature (for AHPs type I) and condensers temperature (for AHPs type II) for the different AHP configurations. The components loads can be estimated in the first steps of the procedure by the fast estimation method proposed by Ziegler & Alefeld (1987), which is presumed to yield COP values with an accuracy of 10% in most cases, sufficient for the purpose of preliminary estimation. Different AHP designs and technical constrains such as maximum or minimum allowable operating temperatures and pressures, and the risk of crystallization should be considered to eliminate some of the potential choices. In the final step of the procedure, a reliable thermodynamic and costing model for AHPs should be used for a high-quality design and cost evaluation.

4.6.3 Interactions Analysis

The temperatures of the AHP components should be estimated considering the interrelation between the process and the AHP. The procedure is illustrated in Figure 5 with two examples; a single stage AHP type I and type II. As mentioned earlier, the maximum evaporator temperature is estimated (solid line in Figure 5a (T_E) and 5b ($T_{E/G}$)) as a function of the target temperature of the HB stream and heat exchanger temperature approach ΔT. Then, the minimum condenser/absorber temperature (for AHP type I) or absorber temperature (for AHP type II) is estimated (square dot line in Figure 5a ($T_{C/A}$) and 5b (T_A)) for each selected heat sink stream and chosen ΔT. The dashed lines in Figure 5a and 5b show the generator temperature (T_G) for a single stage AHP type I and condenser temperature (T_C) for a single stage AHP type II. This phase-diagram mapping should be completed for all selected configurations and working fluid pairs.

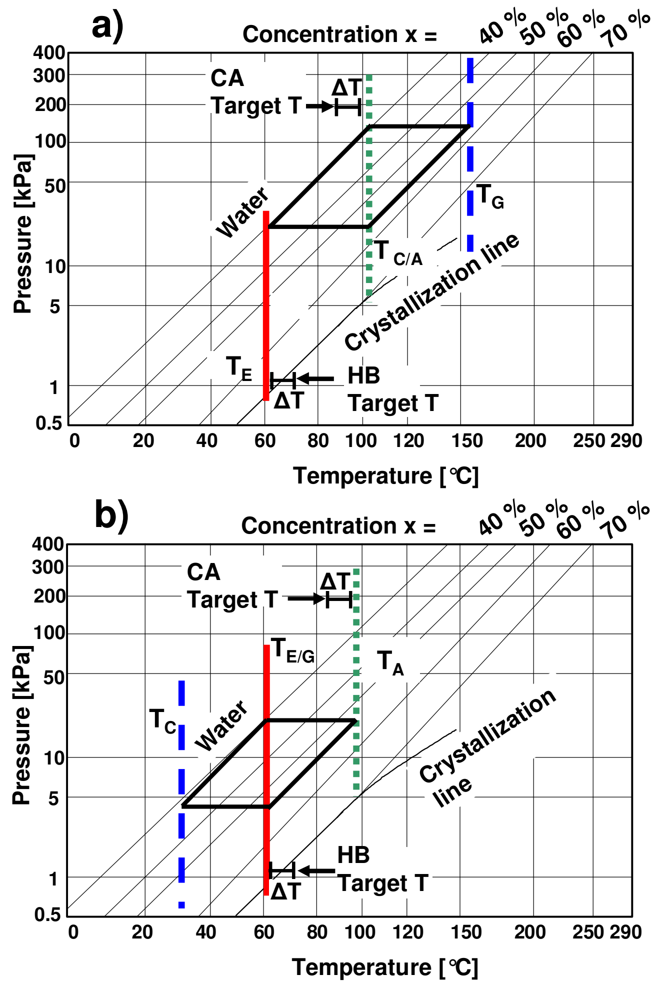


Figure 4-5 Estimation of the maximum and minimum temperatures a) AHP type I b) AHP type II

The controlling stream determines the total amount of useful heat to be produced by the AHP type I, ($Q_A + Q_C$) or by the AHP type II, (Q_A). The capacity of the components releasing heat to the process is subsequently computed by means of the equation for the COP and taking into account the basic relation $Q_G = Q_A + Q_C - Q_E$. It is desirable to have the opportunity stream also as the controlling stream. If it is not the case, one could decide to eliminate that option or consider a combination of streams as the heat source or heat sinks, depending on the situation. From the calculated heat loads and selected temperatures of the generator (for AHP type I) or calculated loads and selected temperatures of the condenser (for AHP type II), and using selected process stream and site utility information, the appropriate streams for the generator (case of AHP type I) or the condenser (case of AHP type II) can be identified. Later, a set of thermodynamically feasible options should be produced by eliminating some of the potential choices, considering both the process and AHP design and technical constraints as discussed above.

A thermodynamic and costing model for AHPs is used for cost evaluation of thermodynamically feasible options. As different economic criteria are used industrially, economic evaluations will be made with three different criteria or combinations there of: pay-back period (PBP), annual net profit and maximum allowed investment cost. Finally, all the practically possible and economically feasible opportunities are identified and can be used in a decision making procedure as to whether the AHP implementation is interesting and detailed design is required. It should be mentioned that the chosen ΔT approach may have an effect on the stream selection and the implementation economics. A sensitivity analysis of ΔT should be done in order to identify the optimal feasible implementation of an AHP in the process.

There are some situations where no practically possible and economically feasible opportunity is identified. In those cases, reconfiguration of the HEN should be considered. The HEN should be modified in such a way as to make other CA or HB streams (which are currently used in the

HEN) available for heat pumping at temperatures closer to the pinch point to avoid large temperature lifts and yield a practical AHP design. In general, the new HEN configuration causes some increase of the total heat exchanger area (capital cost), but the utility demand does not change. In this case, the additional heat exchanger area should be considered in the economic evaluation to determine whether the economic of implementation has been improved by the HEN reconfiguration or not.

4.6.4 Procedure

The procedure for the retrofit of AHPs into manufacturing processes as can be summarized by the following sequence of steps:

1. Selecting the process streams potentially usable by an AHP.
2. Producing a set of different stream combinations (CA/ HB).
3. Estimating the temperatures of the AHP components.
4. Computing all AHP heat loads.
5. Identifying appropriate streams for the generator (type I) and condenser (type II).
6. Producing a set of thermodynamically feasible options.
7. Dimensioning and costing.
8. HEN reconfiguration (if required)

The proposed method introduces several features which reveal superior designs to what could be achieved by either of the two conventional methods for designing AHPs and integration of traditional compression-based heat pumps. Indeed, it considers the interactions between the AHP and the process. It determines the streams that will best fit the AHP, and the best configuration and working pair, taking into account working fluid phase diagrams and other design and technical constrains.

4.7 Case Study

4.7.1 Context

Pulp and paper manufacturing is among the most energy consuming industrial sectors. For example, typical Kraft mill in Canada consumes 25 GJ/ adt[3] of pulp produced and a recently built mill consumes 12 GJ/adt (Browne, 1999). In the Kraft pulping process, the lignin which binds the cellulosic fibers in the wood chips is solubilised by a strong alkaline solution at high temperature and pressure (Smook, 2002). This operation is conducted in reactors called digesters. The spent liquor from the digesting step, the black liquor, is concentrated in a series of evaporators and burned in recovery boilers. In the combustion process, a smelt of sodium sulfide and sodium carbonate is produced and recovered. These chemicals are then recausticized with lime and recycled back to the digester. The cellulosic fibers are separated from the black liquor before it is concentrated in a series of counter-current washers and to form the paper pulp. The pulp is bleached, dried to about 90% solids, cut in sheets and baled for shipment to customers. A simplified schematic of the complete Kraft process is given in Figure 6.

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[3] adt: air dried ton

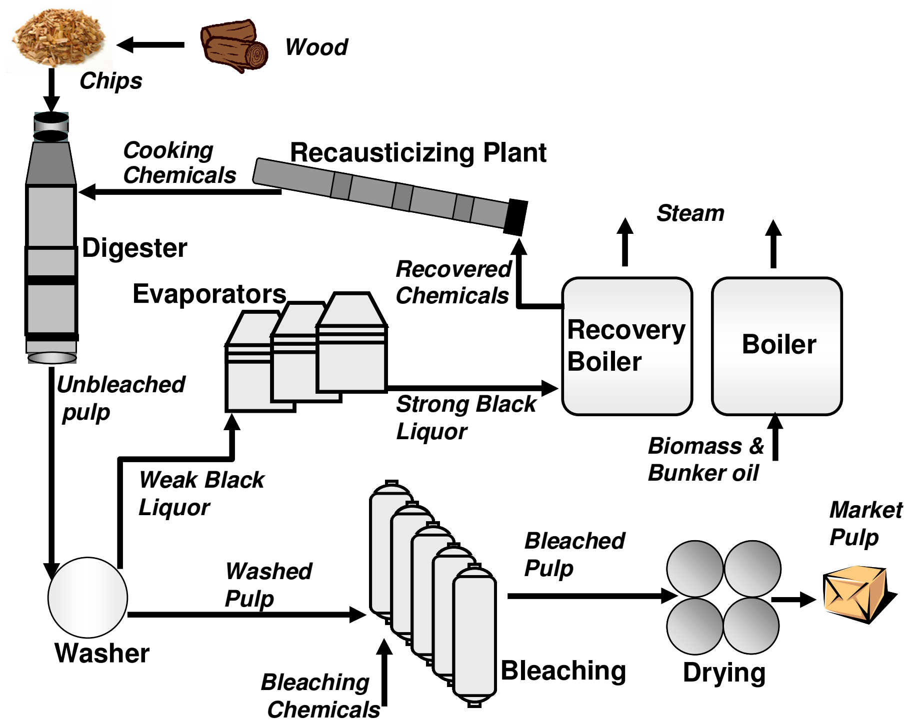

Figure 4-6 Schematic representation of the Kraft process

The mill supporting this study produces 700 ton of high grade bleached Kraft pulp per day. The average steam requirement of the mill is 161 MW (19.1 GJ/adt) produced by four boilers. High pressure steam is produced at 3034 kPa (T = 371 ˚C) by two recovery boilers (64% of total load), a biomass boiler using wood residue (27%) and a bunker oil boiler (9%). About 25 % of the high pressure steam is used directly, 25 % is throttled down to medium pressure (MP) at 965 kPa (178 ˚C) and 50 % to low pressure (LP) at 345 kPa (144 ˚C).

Data were taken from an existing simulation of the mill to develop the composite curves (Figure 4); 61 streams were selected as potential heat sources or sinks (Mateos-Espejel *et al.*, 2008c). A minimum temperature approach ΔT_{min} of 10 ˚C has been chosen, which yields the pinch temperature of 71 ˚C; where hot and cold pinch temperatures are 76 ˚C and 66 ˚C, respectively. The maximum heating and cooling requirements of the process are 178 MW and 65.2 MW while the minimum heating and cooling requirements (MHR and MCR) are 122.8 MW and 10 MW respectively. It was shown in a different study that by maximization of the internal heat recovery, there is an opportunity to reduce the steam consumption of the mill by 30 MW (Mateos-Espejel *et al.*, 2008a).

4.7.2 Background

Several feasibility studies of implementation of AHPs in the P&P industry have been published (Abrahamsson *et al.*, 1994; Costa et al.; Gidner *et al.*, 1996). None of them is based on the results of Pinch Analysis. Since the pinch point temperature, which plays an important role for the correct positioning of AHPs, is not presented and different mills have different pinch points, it is not easy to find if those implementations are well positioned or not. However, it can still be determined for which pinch point ranges they would have been positioned correctly. For example, in the Gidner et al. study (Gidner et al., 1996), an AHP type II is implemented in the evaporation section in which the evaporator, generator, absorber and condenser temperature are 95 ˚C, 69 ˚C, 125 ˚C and 20-25 ˚C; respectively. For the correct positioning of AHP, considering ΔT_{min} of 10 ˚C, the pinch point should have been between 90 and 120 ˚C, which seems high for a Kraft process.

In a previous study, Costa et al. (2009) presented a preliminary feasibility study of the implementation of various AHP configurations in a Kraft pulping process using data from the same manufacturing mill. They have shown that the implementation of AHPs is feasible and cost effective. Since the implementation of the pumps into the process did not observe the principles that are formulated here, in some cases the overall energy benefit is not as assumed in those studies. As an example, a double lift AHP type II was installed to recover heat from the steam flashed at the discharge of the digester. That installation violated the pinch rule by using a stream at 96 ˚C to produce LP steam, therefore affecting a transfer of heat loads rather than a reduction of steam demand. The present case study shows how the proposed method can be used to avoid such errors and correctly position an appropriately designed AHP in the process.

4.7.3 Procedure

Step 1. Table 1 shows the selected streams that are available heat sources and sinks in the process after maximization of the internal heat recovery. HB1 stands out as an interesting heat source; its temperature (75.9 °C) is just below the hot pinch temperature and it carries a high load (12.2 MW). There are 5 heat sinks available, CA1 to CA5. However it should be noted that because of high pulp concentration, CA1 and CA2 will not be considered due to the difficulties of handling such concentrated streams.

Table 4-1 Selected streams

	Heat sinks above the pinch point (CA) Section	Description	State (C or H)	T_{start} (°C)	T_{target} (°C)	Flow (kg/s)	Heat load (kW)
1	Bleaching	Pulp through vapormixer 3	C	68.6	75.0	58.8	1471
2	Bleaching	Pulp through vapormixer 4	C	71.9	85.0	51.7	2626
3	Evap. #3	Liquor through HXC	C	104.9	132.2	14.0	8907
4	Evap. #2	Liquor through HXC	C	101.4	127.8	14.5	7363
5	Dearator	Condensate and fresh water	C	66	100	67.6	9653
	Heat sinks below the pinch point (CB)						
1		Fresh water from 4 °C (winter) or 20 °C (summer) to any temperature below the pinch point.					
	Heat sources above the pinch point (HA) section	Description	State (C or H)	T_{start} (°C)	T_{target} (°C)	Flow (kg/s)	Heat load (kW)
1	Boilers	Flue gas from CR2	H	164.0	105.8	63.5	4314
2	Boilers	Flue gas from CR3	H	199.0	117.6	47.6	4625
3	Boilers	Flue gas from bark boiler	H	182.0	126.5	47.7	2863
	Heat sources below the pinch point (HB) section	Description	State (C or H)	T_{start} (°C)	T_{target} (°C)	Flow (kg/s)	Heat load (kW)
1	Evap. #2	Vapor condensing	H	75.9	74.9	5.3	12211
2	Bleaching	Effluent, washer 1	H	58.1	33.0	58.8	6150
3	Bleaching	Effluent, washer 2	H	68.6	33.0	67.0	9911
4	Bleaching	Effluent, washer 3	H	68.2	33.0	18.8	2768
5	Bleaching	Effluent, washer 5	H	66.8	33.0	9.5	1338
6	Bleaching	Effluent washer 4	H	73.1	33.0	36.0	6027

Step 2. As a result, the combination map is HB1& CA3, HB1& CA4 and HB1& CA5.

Step 3. In the next step the temperatures are estimated. Considering the temperature of the heat source stream (75.9 °C) and ΔT=10 °C, the maximum evaporator temperature should be 65.9 °C. The minimum condenser/absorber temperature or absorber temperature depending upon the type of heat pump that may be utilized (AHP type I and type II) are estimated as 142 °C, 138 °C and 110 °C for CA3, CA4 and CA5 respectively. In this example 6 different configurations (single stage, double effect and double lift AHP type I or type II) for the two common working fluid pairs (LiBr-H_2O and NH_3-H_2O) are considered; this gives 36 different combinations. Table 2 shows the thermodynamically feasible configurations for the resulting combinations. Those that are not presented in the table are too remote from feasible conditions; some would lead to excessive generator temperatures for an AHP type I, some to very low condenser temperatures for an AHP type II others would fall in the crystallization zone. The estimated generator temperature for the AHP type I cases and condenser temperatures for the AHP type II cases are presented for the 20 remaining feasible configurations.

Table 4-2 Thermodynamically feasible configurations

Case	H.Si * stream	Confi-guration		Working pair	COP	Q_G (MW)	Q_E (MW)	Q_A (MW)	Q_C (MW)	T_C (°C)	Low T H.Si	T_G (°C)	High T H.So
1	CA3	Type I	SS	H2O-LiBr	1.72	5.17	3.73	5.17	3.73			250	HP
2			DL	H2O-LiBr	1.3	6.85	2.05	3.97	4.93			190	HP
3				NH3-H2O	1.23	7.24	1.66	4.41	4.49			200	HP
4		Type II	SS	NH3-H2O	0.42	5.12	7.08	5.12	7.08	10	NF		
5			DL	H2O-LiBr	0.29	7.44	4.76	3.54	8.66	30	FW		
6				NH3-H2O	0.27	5.25	6.95	3.29	8.91	30	FW		
7	CA4	Type I	SS	H2O-LiBr	1.72	4.3	3.1	4.3	3.1			220	HP
8			DL	H2O-LiBr	1.3	5.69	1.71	3.3	4.1			190	HP
9				NH3-H2O	1.23	6.02	1.38	3.66	3.74			190	HP
10		Type II	SS	NH3-H2O	0.42	5.12	7.08	5.12	7.08	10	NF		
11			DL	H2O-LiBr	0.29	7.44	4.76	3.54	8.66	35	FW		
12				NH3-H2O	0.27	5.25	6.95	3.29	8.91	30	FW		
13	CA5	Type I	SS	H2O-LiBr	1.72	5.61	4.04	5.61	4.04			160	HP
14				NH3-H2O	1.55	6.23	3.42	6.23	3.42			170	HP
15			DL	H2O-LiBr	1.3	7.41	2.23	4.31	5.34			140	HP-MP-FG
16				NH3-H2O	1.23	7.85	1.8	4.78	4.87			150	HP-MP-FG
17		Type II	SS	H2O-LiBr	0.47	5.73	6.5	5.73	6.5	30	FW		
18				NH3-H2O	0.42	5.12	7.08	5.12	7.08	25	FW		
19			DL	H2O-LiBr	0.29	7.44	4.76	3.54	8.66	30	FW		
20				NH3-H2O	0.27	5.25	6.95	3.29	8.91	30	FW		

* H.Si: Heat Sink; H.So: Heat Source; SS: Single Stage; DL: Double Lift; HP: High pressure steam; MP: Medium Pressure steam; NF: Not Feasible; FW: Fresh Water; FG: Flue Gas.

Step 4. Once the COP value has been estimated by the Ziegler & Alefeld method (Ziegler and Alefeld, 1987), the load for each component can be computed and their values are given in Table 2 (Q's). For all AHPs type II cases, fresh water could be used as low temperature heat sink; except in cases 4 and 10 where the condenser temperature is too low and that will not be considered. For all AHPs type I, high pressure steam could be used as the high temperature driving energy. For some of them, medium pressure steam could also be used (cases 15-16) and for others, part of the required heat load could come from flue gases (cases 15-16) (step 5).

Step 5. Step 5 consists of eliminating additional combinations by considering design and operating constraints. Some of the cases are clearly impractical and are readily eliminated. Considering the corresponding working pair equilibrium diagram; cases 1, 2, 3, 7, 8 and 9 are eliminated because of the high generator temperature (they have generator temperature at 190 °C and above), case 4 and 10 are eliminated because 10 °C would be required for the condenser temperature. Cases 1 and 7 could also be eliminated because of the high risk for crystallization and Case 14 is eliminated because of the high operating pressure of its cycle.

Figures 7a & 7b illustrate why case 1 and case 10 are eliminated. In Figure 7a, the condenser/absorber and evaporator temperatures are used to illustrate the thermodynamic cycle of case 1. The estimated generator is clearly too high for current AHP technology (250 °C). Also, because of the high LiBr concentration in the solution side, there is a potential risk of crystallization during operation. Figure 7b shows that case 10 is eliminated because of the low condenser temperature. Reaching a temperature of 10 °C in the condenser with a ΔT of 10 °C, would entail a heat sink at or be below 0 °C, which is not practical. It should be mentioned that it has been verified by observation of the site mill layout that all selected streams are relatively close to each other and will not require extensive piping work. Another reasonable constraint to be considered to eliminate additional cases is the fact that as much as possible of the available energy of the selected heat source (12.2 MW) should be upgraded. Cases 13, 15 and 16 are thus eliminated, because they use only 33, 18 and 15 % of the available 12.2 MW, respectively. At the end of this elimination procedure, 8 cases (cases 5, 6, 11, 12, 17, 18, 19, 20) remain from the 36 identified originally; they are all single stage or double lift AHPs type II. They will now be compared on the basis of economic and thermodynamic criteria for the final choice.

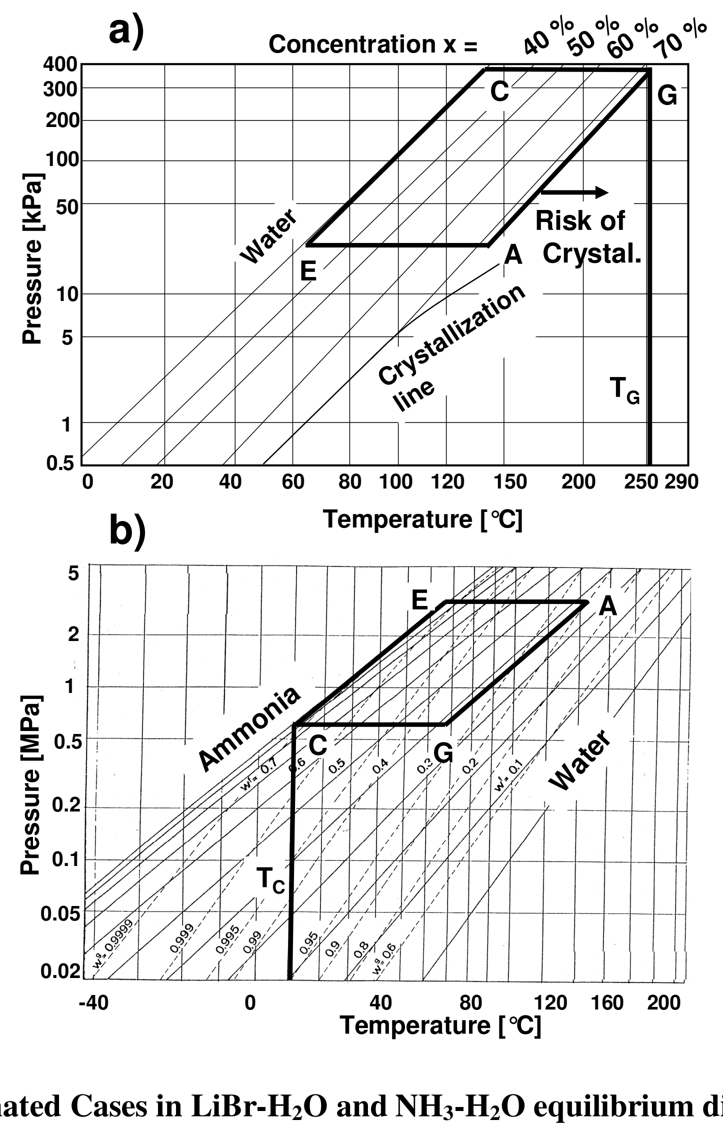

Figure 4-7 Eliminated Cases in LiBr-H_2O and NH_3-H_2O equilibrium diagram; a) case 1; High generator T and risk of crystallization b) case 10; low condenser T

Step 6. Case 17, which has the highest estimated delivered heat (5.73 MW) as a single stage AHP type II, and case 19, which has the highest estimated delivered heat (3.54 MW) among the double lift AHPs type II, are chosen to be dimensioned and cost estimated, as they are likely to be the most attractive options.

Step 7. For this purpose, a simulation model for the single stage and the double lift AHP type II using the LiBr/H_2O working pair was used (Bakhtiari, 2009). It calculates the internal mass and energy balances of each component as well as the heat transfer between external and internal streams in a steady state operation. The parameters of a cycle such as output temperatures, pressure of each components, heat load of heat each exchanger and COP are calculated from input values such as input temperatures and mass flow rates. The results from this simulation were validated with data from the literature (Herold et al., 1996). Table 3 presents the basic design parameters and simulation results. The useful delivered energy in the absorber is calculated at 5.83 and 3.89 MW for case 17 and case 19. The power to be supplied to the generator and evaporator were then calculated at 5.76 and 6.55 MW for case 17 and 7.45 and 4.76 MW for case 19. The COP is also calculated as 0.48 and 0.32 for case 17 and case 19. Figure 8 shows the feasible cycles in the LiBr/H_2O equilibrium diagram.

Table 4-3 Basic design parameters for selected cases

Variable	Unit	E	G	A	SHX	C	E1-A1
Case 17 (SS AHP type II)							
LMTD	°C	9.3	10.3	17.3	10.9	12.8	
Pressure	kPa	26.3	3.9	26.3	26.3	3.9	
U.A.	kW/°C	693.5	559.2	337.0	100.0	498.4	
Q	MW	6.5	5.8	5.8	1.1	6.4	
Case 19 (DL AHP type II)							
LMTD	°C	11.2	11.6	22.9	14.5	8.8	4.2
Pressure	kPa	24.2	4.3	33.3	24.2	4.3	24.2-33.3
U.A.	kW/°C	425.0	642.2	169.9	93.0	945.5	904.8
Q	MW	4.8	7.5	3.9	1.3	8.3	3.8

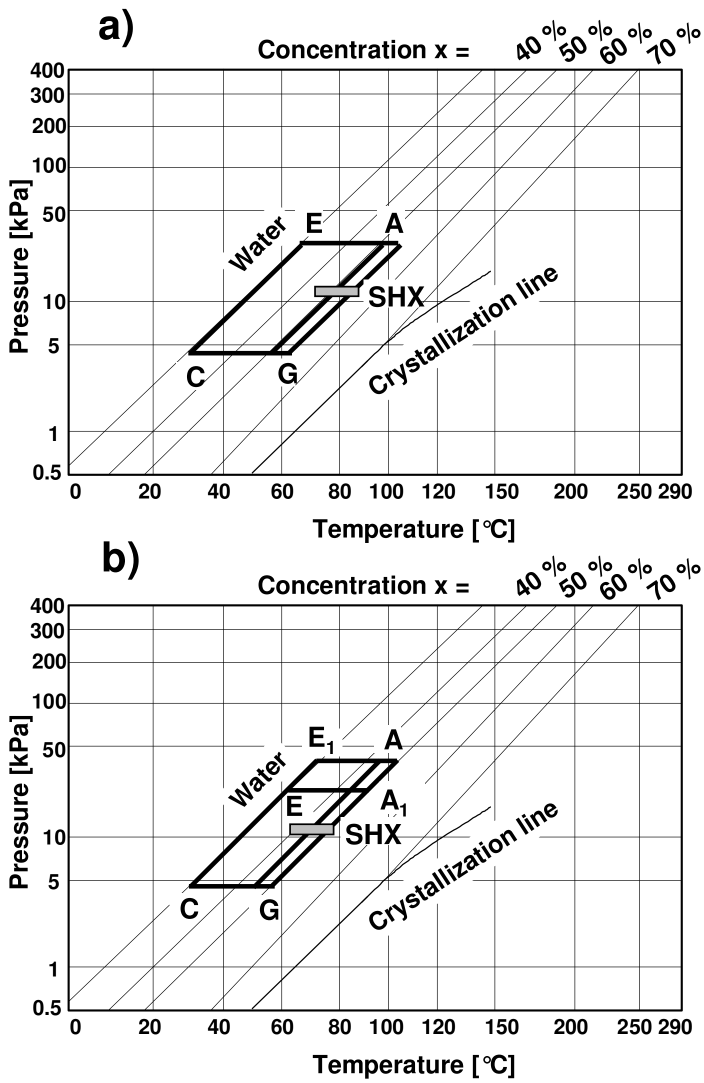

Figure 4-8 The selected cycles in LiBr/H_2O equilibrium diagram; a) Case 17; single stage AHP type II b) Case 19; double lift AHP type II

The economic feasibility of those two AHP implementations was assessed. The installed cost of each device was estimated on the basis of output heat in $/kW. The following cost compiled by the US DOE (RCG/ Hagler-Bailly, 1990) were used[4], 581 and 656 $/kW for single stage and double lift, respectively (those costs are almost identical to those presented by Berntsson et al. (Berntsson and Frank, 1997)). The annual rate of operations, steam and cooling water costs were taken as 8640 hours, 62.5 $/MWh and 1 $/MWh, respectively as suggested by mill personnel. Maintenance and other operating costs were neglected. The estimated installed cost of the AHPs and the annual saving in steam and cooling water are presented in Table 4. The simple payback times (SPB) given by Equation 1 are 1 and 1.2 years for the single stage and double lift AHP type II.

$$\text{SPB} = \text{Investment} / \text{Yearly Savings} \qquad (1)$$

Table 4-4 Economic evaluation of selected cycles

Case	SS AHP type II (Case 17)	DL AHP type II (Case 19)
Installed Cost [M$]	3.4	2.6
Steam saving [M$/a]	3.2	2.1
Cooling saving [M$/a]	0.11	0.11
SPB	1	1.2

Life-cycle costs of the two case studies were compared using the net present value (NPV) (Costa et al.). The initial investment cost and the yearly energy savings were estimated over the life cycle, which was set at 15 years. A discount rate of 7%, which is usual for energy projects and a

[4] All costs in this paper are given in 2008 Canadian dollars

yearly escalation rate of 4% for fuel price have been used for the time span of the investment. Figure 9 shows the evolution of the NPV over 15 years; the abscissa gives the time elapsed from the date of investment and the intersection with the nil value line indicates the time required for the projects to become cost-effective. It is about one year in the two cases for the current steam price. The figure shows that the two options are interesting not only in the short term (quite short PBT), but also in the long term. The initial investment for case 19 is more attractive although both cases produce almost the same SPB. However, case 17 which produces high yearly revenue perform best in the long term, even if the initial investment is higher. It should be noted that neither operating costs nor potential GHG emission credits were taken into account. The latter could be a significant factor of payback time reduction.

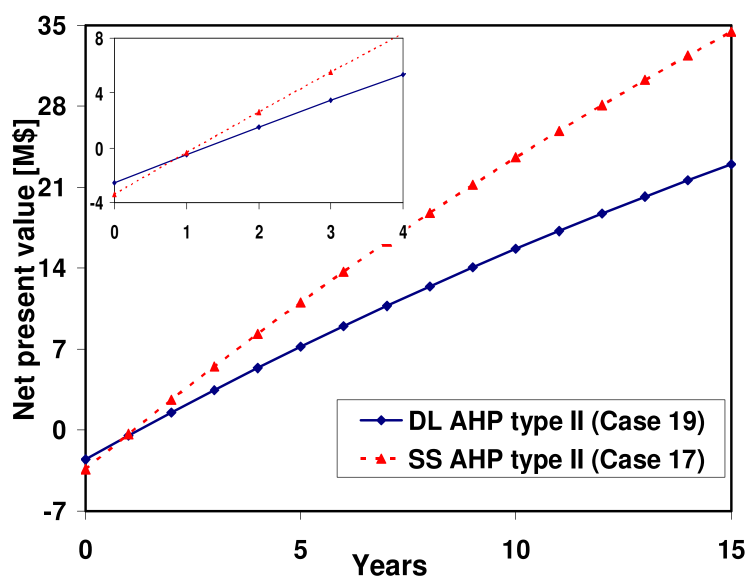

Figure 4-9 Net present value for the two case studies

Step 8. Since two practically possible and economically feasible configurations have been already identified, the HEN reconfiguration is not considered in this case study (step 8). The authors presented another study in which the HEN reconfiguration and the effect of water closure on the pinch point are also presented (Bakhtiari *et al.*, 2009).

4.8 Conclusion

The implementation of an AHP in a process should be supported by a complete process analysis respecting both process and machine constraints. The opportunities for correct positioning of AHPs in a process were identified and guidelines were formulated for its appropriate positioning in the process. Results from a case study based on a real plant has validated the method and illustrated its usefulness.

In the proposed method, cases with different configurations and different working fluid pairs are compared economically with each other. At the present time, there is no reliable cost estimation model for AHPs, considering different configuration and working pairs and this work has shown that there is a need for the developement of such a tool.

One important conclusion from the case study is that even for a heat-integrated process, there can still be room for additional utility savings. For the selected potential heat source, 8 different thermodynamically and technically feasible configurations were identified. It was found that a single stage and a double lift AHPs of type II were the most realistic implementations and their thermal design and preliminary economic evaluation is presented. Simple pay back time is about one year for the current steam price for the two cases. The NPV has also been computed and has shown that the two options are also interesting in the long term. The single stage AHP type II is the best configuration for the actual case considered; because of almost the same SPB, better performance in the long term and much simpler design.

4.9 Acknowledgments

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CHAPTER 5 - OPPORTUNITIES FOR THE INTEGRATION OF ABSORPTION HEAT PUMPS IN THE PULP AND PAPER PROCESS

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Keywords: Absorption heat pumps, pulp and paper industry, Kraft process, Heat upgrading.

5.1 Presentation of the Article

In this article, opportunities for the integration of absorption heat pumps in the pulp and paper process as the second specific objective of this research are presented. The developed methodology is used to identify generic opportunities for the implementation of AHPs in the Kraft process. This article is submitted to Energy journal.

5.2 Abstract

Implementation of absorption heat pumps (AHP) in a Kraft pulping process was studied using a new methodology for the optimal integration of those devices in a process. Two generic opportunities were identified for an energy and water optimized mill: (i) integration of a double lift chiller in the bleaching chemical making plant to produce chilled and hot water simultaneously, using MP steam as the driving energy and, (ii) installation of a single stage heat pump to concentrate the black liquor and produce useful hot water by upgrading heat from the bleaching effluent and using MP steam as driving energy. The principles of AHPs operation and

their efficient integration into a process are described. The simple payback time (SPB) and net present value (NPV) were used to evaluate the interest of such implementations. Considering 63 $/MWh for the steam price, SPB of 2.7 and 1.7 years have been estimated for the two cases.

5.3 Introduction

Pulp and paper (P&P) manufacturing is among the most energy intensive industrial sector. For example, an old Kraft mill in Canada consumes 25 GJ/ adt (air dried ton) of pulp produced while a modern mill consumes 12.2 GJ/adt [1]. Over the years, this industry has invested much effort to reduce its energy bill and significant progress has been made by the application of a broad spectrum of energy enhancing measures [2]. Unfortunately considerable amount of energy is still rejected to the environment because of its low temperature. Heat pumps (HP) could be used to upgrade this heat to useful temperature levels. HPs are energy conversion devices which are used to upgrade the quality of heat by raising the temperature at which it is available [3]. Absorption heat pumps are emerging as a potential alternative to the more common vapor recompression heat pumps (VRHP). They are thermally driven and when judiciously positioned into an industrial process, they can be operated with practically no purchased power. They also use environmentally benign working fluids.

To our knowledge, there are only few reports of actual implementation of AHPs in the P&P industry in the scientific literature. The first documented AHP unit installed in a P&P mill has been in operation for 15 years [4]. It is a 200 kW heat transformer that delivers steam at 125 °C using secondary vapor at 85 °C from the liquor evaporation plant. At a pulp factory in Japan, an AHP was installed to recover heat from the alkaline bleaching effluents, at about 50

°C and used to increase the temperature of water supplied to the boiler from 22 to 80 °C by using 1.05 MW of steam (0.4 MPa, 165 °C) [5]. Several Experimental developments and feasibility studies have been reported. Hester et al. [6] present a conceptual design and economic evaluation of a heat recovery/heat pump system as an integral part of a pulp and paper plant. Robb et al. [7] present a feasibility study for upgrading heat rejected by newsprint processes to displace fossil fuel generally used to dry the product. Abrahamsson et al. [8] present two different potential applications of AHPs in the P&P industry. In the first application, optimal energy conservation strategies were investigated using a heat transformer system incorporated with the evaporation plant of the pulping process. In the second application, simulation results were presented for different process configurations where an absorption heat pump was suggested to be incorporated in an existing paper drying plant. Costa et al. [9] present two feasibility studies of implementation of AHPs in a Kraft process. One involved the insertion of a heat transformer in the heat recovery loop from the digesters blow-down tanks; the second presented the installation of a chiller to replace the barometric condenser in the bleaching chemical making plant.

In none of these reported feasibility studies is Pinch Analysis [10] taken into account. As will be discussed in this paper, it can easily be shown that when the HP is not positioned correctly, the net benefits are practically nil. The methodology of integration of some types of HPs, such as VRHP or electrically driven compression HPs, is well defined and discussed in the literature [11, 12]. Recently, Bakhtiari et al. [13] presented a new methodology for process integration of AHPs. It will be summarized in this paper and used to identify generic opportunities in a Kraft process.

The work presented here illustrates the potential use of AHPs for heat upgrading in the P&P industry. It shows that even for a fully energy and water optimized mill, there is still a potential for further utility savings. Two generic opportunities have been identified that involve a broad spectrum of machine configurations, mode of operation and integration context. To provide background for this case study, the principles and operation of heat pumps are first briefly reviewed and a method based on Pinch Analysis to appropriately position an AHP in a process is presented.

5.4 Absorption Heat Pumps

Much scientific and technical literature has been devoted to the fundamental principles, modes of operation and engineering design of HPs [3, 14, 15]. A heat pump is a mechanical device able to raise the temperature at which a certain quantity of heat is available to a higher temperature where it can be used more advantageously. This is accomplished by circulating a working fluid (called the refrigerant) between an evaporator, E, and a condenser, C, which is operated at higher temperature and pressure (Figure 5-1a). The circulation of the working fluid between the evaporator and the condenser is caused by a pressure raising device (PRD), which is driven by high quality energy. In the case of the AHP, the PRD consists of a secondary loop (Figure 5-1b) where a binary solution is circulated between an absorber, A, operating at the same pressure as the evaporator and a desorber, generally called the generator, G, at the same pressure as the condenser. The refrigerant is the more volatile component of the binary solution; the second component is the solvent. The absorption process involves an exothermic reaction where the vaporized refrigerant exiting the evaporator is absorbed by the solvent strong solution releasing additional useful heat, Q_A. The solvent weak solution thus formed is pumped up to the generator and released at higher pressure and temperature under the effect of driving energy supplied as heat, Q_G. In an actual machine a fifth heat exchanger, the solution heat exchanger (SHX) is inserted between the generator and the absorber to enhance the cycle efficiency. AHPs are generally used to upgrade the temperature level of some available heat at a low temperature level (Q_E) to recover useful heat at an intermediate temperature (Q_C+Q_A) using the high temperature heat source as the driving energy (Q_G). In the majority of AHPs developed for use in industrial applications, NH_3/H_2O and H_2O/LiBr have been the working fluid pairs of choice. Other working fluid pairs are also reviewed in the literature [13].

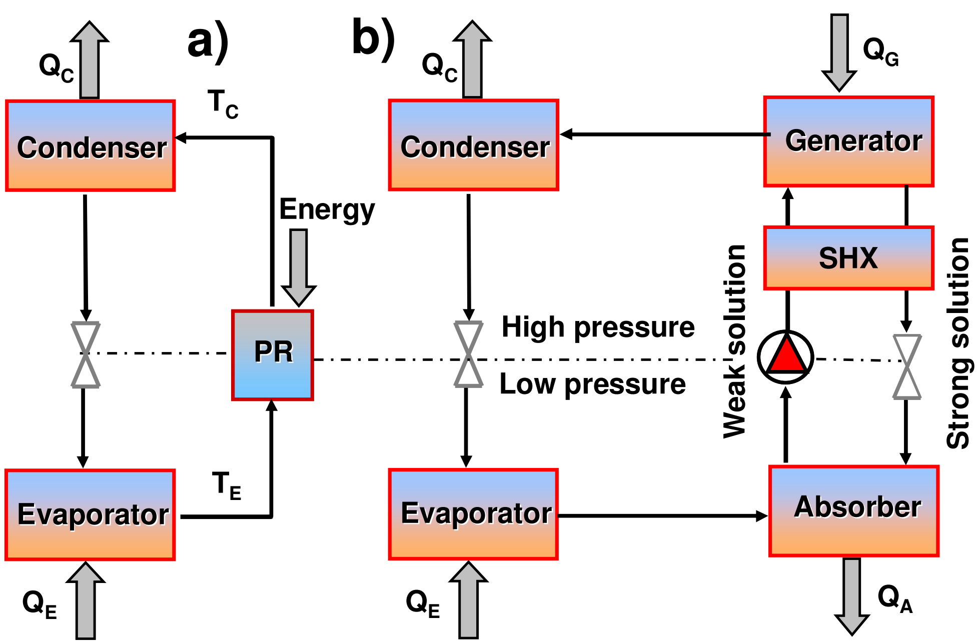

Figure 5-1 Principle of heat pumping; a) General; b) AHP

For preliminary investigations, comparisons of working fluids and thermal analysis of an AHP, the pressure-temperature diagram (or vapour pressure diagram as it is often referred to) is a very useful tool. It is convenient to represent the cycle in such a phase diagram of the binary solution (Figure 5-2) illustrating the thermal operating constraints imposed on the system, the pure refrigerant evaporation line on the low temperature side and the solvent crystallization line on the high temperature side. A schematic representation of the configuration of AHPs based on the pressure-temperature diagram is often used; Figure 5-3a is such a symbolic representation of an AHP. AHPs can be implemented in a variety of arrangements to overcome the limitations imposed by the thermodynamics and the irreversible effects associated with heat transfer. Of interest to this work is one other configuration, the double lift AHP in which the higher temperature lift can be achieved by coupling cycles (Figure 5-3b).

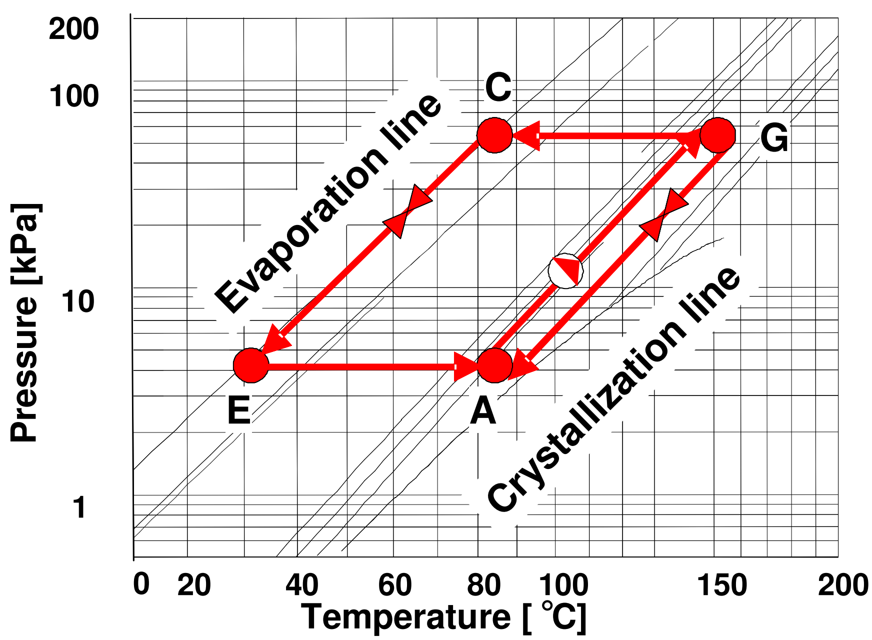

Figure 5-2 AHP representations in H_2O/LiBr phase diagram

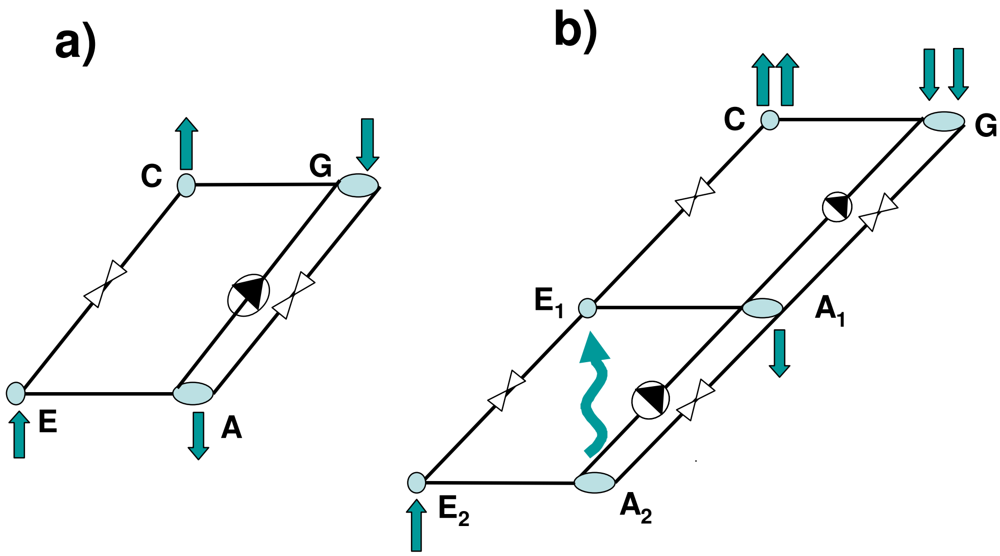

Figure 5-3 Configurations in P vs T representation; a) single stage AHP, b) double lift AHP

The efficiency of an AHP or its coefficient of performance (COP) is the ratio of the energy released by the machine to the driving energy as follows: COP = ($Q_C + Q_A$) / (Q_G).

5.5 Appropriate Positioning of Absorption Heat Pumps in a Process

This section, presents how an AHP should be integrated in industrial processes. It has been shown that a heat pump should be implemented in the process based on the results of Pinch Analysis [11-13]. Pinch Analysis is a technique used to maximize internal heat recovery within a process [16]. The thermodynamics of Pinch Analysis dictates a fundamental rule: there must not be transfer of heat across the pinch point. If this happens, the process will suffer a double penalty, the simultaneous increase of the cooling and heating demands. Positioning of heat pumps is an exception to this rule.

Bakhtiari et al. [13] present how the composite curve (CC) diagrams can be used to position AHPs in the process as illustrated on Figure 5-4. The condenser and absorber must release their heat above the pinch point to reduce the minimum heating requirement (MHR); the generator which is at a higher temperature can thus only be above the pinch point and, to reduce the minimum cooling requirement (MCR), the evaporator must be below. It was shown that the overall energy gain is effectively the same as for a VRHP of equal power, but the need for shaft work is eliminated. They have developed a general methodology for the process integration of AHPs. Their developed methodology is summarized in the organigramme of Figure 5-5. The methodology relies on data extracted from a Pinch Analysis of the plant. That methodology provides systematic guidelines for the proper selection of heat sources and sinks that will maximize the benefit derived from heat pumping, while respecting both process constraints and operating requirements of the AHP.

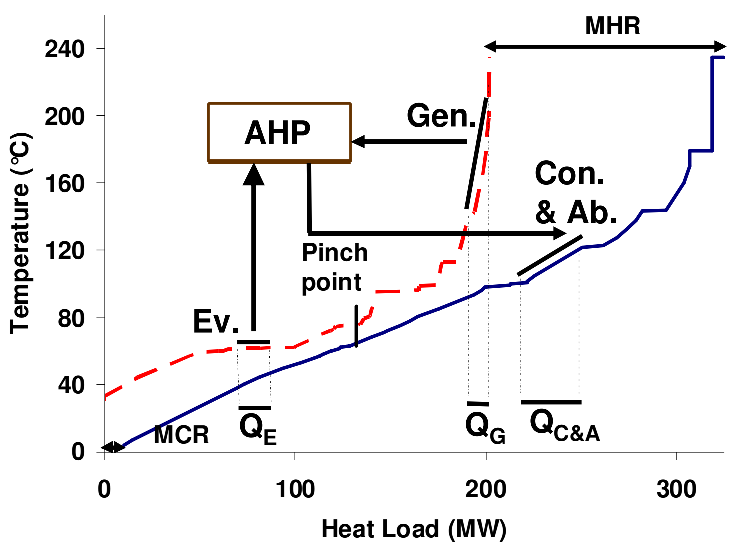

Figure 5-4 Appropriate positioning of an AHP

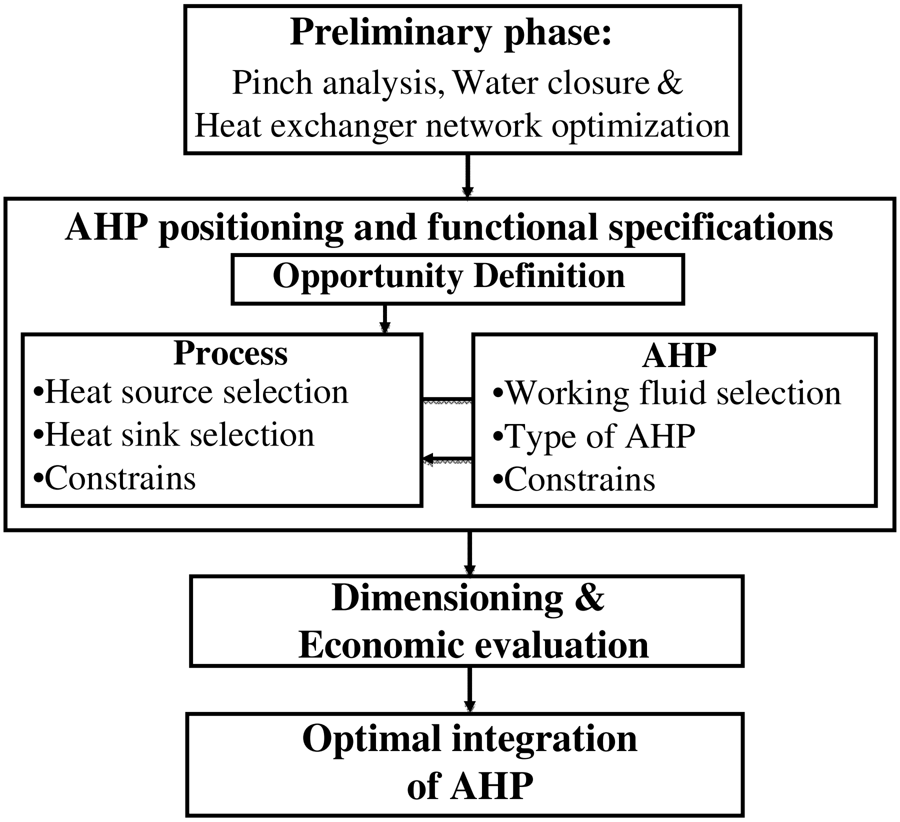

Figure 5-5 Methodology of integration of AHPs [13]

None of the published feasibility studies of implementation of AHPs in the P&P industry are based on the Pinch Analysis. Since the pinch point temperature plays an important role for correct positioning of AHPs and different mills have different pinch points, it is not possible to determine weather correct positioning was used in those published studies. However, the range of pinch point temperatures for which AHP positioning would be correct can be ascertained. For example in the Abrahamsson et al study [8], the generator temperature is 140 ˚C, condenser and absorber operate at 80 ˚C and evaporator operates at 50 ˚C, so for correct positioning of an AHP, the pinch point should have been between 50 and 80 ˚C, which is reasonable for a P&P mill [17, 18].

5.6 The Reference Kraft Mill

The Kraft process is the predominant pulping method by which wood chips are transformed into paper pulp [19]. Figure 5-6 shows a simplified schematic of the complete Kraft process. As a basis for the comparison, the energy consumption for a model Kraft mill is presented in [1]. It is shown that the model mill consumes 12.2 GJ/adt and the largest energy consumers in the mill are: bleaching, pulp machine, black liquor evaporators and power plant. To present actual opportunities for the process integration of AHPs in the Kraft process, the data utilized was taken from a Kraft pulp manufacturing mill currently in operation in eastern Canada.

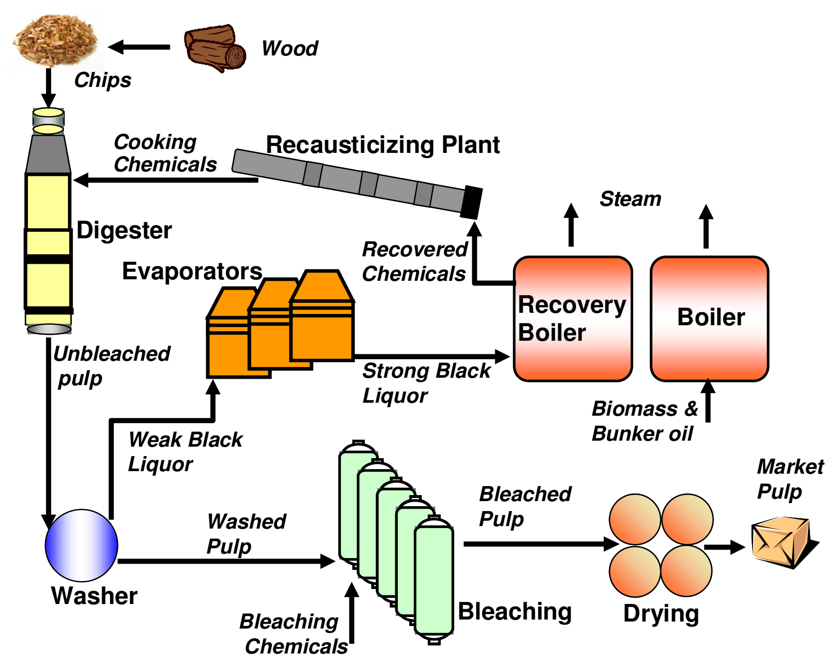

Figure 5-6 Schematic of Kraft process

The studied mill produces 700 adt/d of high grade bleached Kraft pulp. The average steam requirement of the mill is 19.1 GJ/adt (161 MW) produced as high pressure steam (HP) at 3034 kPa (T = 371 ˚C) by four boilers: two spent liquor recovery boilers (64%), a biomass boiler using wood residue (27%) and a bunker oil boiler (9%). About 25 % of the HP steam is used directly, 25 % is throttled down to medium pressure (MP) at 965 kPa (178 ˚C) and 50 % to low pressure

(LP) at 345 kPa (144 ˚C). The average water consumption of the mill varies from 83.5 m^3/adt during the winter to 102 m^3/adt in summer.

Supporting data for this study were taken from an existing simulation of the mill on CADSIM Plus to develop the CC (Figure 5-4); 61 streams were selected as potential heat sources or sinks [20]. The actual heating and cooling requirements of the process are 178 MW and 65.2 MW. It was shown that after water closure and condensate return, the heating and cooling demand of the process is reduced by 14 MW and 4.2 MW respectively (Mateos-Espejel et al., 2008b). A minimum temperature approach, ΔTmin, of 10 ˚C was chosen, which yields a pinch temperature of 57 ˚C and minimum heating and cooling requirements (MHR and MCR) of 96 MW and 22 MW respectively. It was also shown that steam consumption can be further reduced by 21 MW and the cooling requirement by 20 MW by optimizing the heat exchanger network to maximize internal heat recovery. It is worth mentioning that water closure and condensate return should be performed before Pinch Analysis and heat exchanger network optimization. Normally this will reduce the MHR and the pinch point temperature and increase the MCR of the process. In general, decreasing the pinch point temperature by reducing the required temperature lift facilitates the implementation of AHPs.

In the studied Kraft process, there is a very large amount of energy at low temperatures which has the potential to be recovered. Under current process configuration, much of that energy is wasted. The bleaching plant (bleaching effluents and chilled water production) is among those plants with the greatest potential for heat upgrading. Analysing the optimized heat exchanger network [17], it is observed that most of the available heat sinks above the pinch point are at relatively high temperature levels. To avoid the required large temperature lift and to have practical AHP design, the optimized heat exchanger network was reconfigured. The new configuration caused the available heat sinks to be at temperatures closer to the pinch point. It caused some increase of the total heat exchanger area (capital cost), but the utility demand did not change. Heating and cooling requirements will be reduced further by the installation of AHPs.

5.7 Opportunity I

The first identified opportunity deals with the production of chilled water at 6 °C in the chlorine dioxide (ClO_2) making plant. Modern bleaching is achieved through continuous sequences of process stages utilizing different chemicals and conditions in each stage, usually with washing between stages. Different chemicals are used in each stage; such as: NaOH, ClO_2, oxygen, hypochlorite, peroxide and ozone. ClO_2 is a gas at room temperature/pressure, and liquefies at 11 °C. However it is unstable and potentially explosive at room temperature. In air mixtures, it is easily detonated by exposure to heat, light, mercury and various organic substances. ClO_2 is somewhat more soluble in water than chlorine (10 to 11 grams per liter at 4 °C). Since it cannot be shipped either in pure form or as a concentrated solution, ClO_2 is always manufactured at the mill site. It is generated as a gas from chemical reduction of sodium chlorate in a high acidic solution; the gases then absorbed in cold water to produce ClO_2 solution at a concentration of about 7 g/liter. In the current process configuration of the mill, shown in Figure 5-7, chilling water is accomplished by a commercial compression heat pump requiring 260 kW of shaft work and, by a steam ejector, which is operated with MP steam supplied at a rate of 0.6 kg/s. The total chilled water production is 23.3 kg/s and to cool both units, 139.5 kg/s of cooling water are required. In a previous study of the same mill [9], the steam ejector was replaced by a double effect absorption chiller driven by MP steam and using the H_2O/LiBr working pair. With a cooling COP of 1.25, the chiller would require only 0.125 kg/s of MP steam (349 kW) for the same chilled water production. Although that implementation seems promising, it can be even more interesting if the AHP is positioned correctly using the developed methodology and made to deliver the energy above the pinch point.

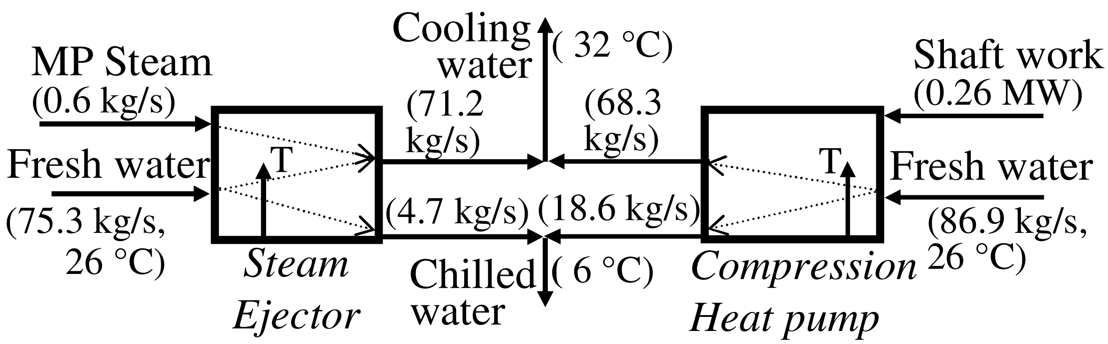

Figure 5-7 Flow diagram of current chilled water production in the chemical making plant

Considering the developed methodology (Figure 5-5), the process is first fully thermally optimized with the premise that there will be an AHP implementation as the following step. In this case the objective is chilled water production, which means there is a heat source below the pinch point which has the potential to be upgraded to above the pinch point. Table 5-1 shows a list of available heat sinks above the pinch point. Since the heat source load (23.32 kg/s, 26/6 °C; 1.95 MW) is relatively high, a combination of heat sink streams above the pinch point should be selected. Considering the temperature of the heat source stream (from 26 to 6 °C) and ΔT=4 °C (a reasonable low ΔT because of the low temperature), the maximum evaporator temperature is estimated at 2 °C. In this example 6 different configurations (single stage, double effect and double lift AHP or AHT) for the two common working fluid pairs (LiBr-H_2O and NH_3-H_2O) are considered; this gives 36 different combinations. Considering the working pairs phase diagrams and all different AHP configurations, all AHTs that lead to very low evaporator temperatures are eliminated from the possibilities. Also, since the energy should be delivered to above the pinch point and a high temperature lift is needed, a stand alone single stage AHP and double effect cycles can not be applied. The only two remaining possibilities are a double lift AHP using LiBr-H_2O and the same system using NH_3-H_2O as the working pair. Since the NH_3-H_2O system leads to work at very high pressure, it is also discarded from the list. Once the COP value has been estimated from Ziegler & Alefeled value [21], the load for each component of the double lift AHP using LiBr-H_2O can be computed. It is found that the system would deliver around 8.4 MW heat above the pinch point which will satisfy streams 1, 2 and a part of stream 3 from Table 5-1. Considering a temperature approach of 10 °C for the condenser and generator, their minimum temperature is estimated at 85 and 150 °C, respectively. For this purpose, a simulation model for the single stage and double lift AHP using the LiBr/H_2O working pair was developed. This model calculates the output parameters (outlet temperatures, solution concentrations and pressures) of cycle during a steady state operation. Knowing the output temperatures, the heat load for each heat exchanger and the COP are calculated. Results from this simulation were validated with data from the literature [3]. Table 5-2 presents the basic design parameters and simulation results for the proposed double lift AHP. Figure 5-8 shows that the desired temperature lift can be accomplished in a double lift machine. Figure 5-9 presents the proposed flow diagram of chilled water production plant. Chilled water is produced at the evaporator 2 at a rate of 84 t/h, using 3.13 kg/s of MP steam as the driving energy in the generator. Condenser and absorber 1 release the useful energy above the pinch point to increase the temperature of 122 kg/s of the hot water in the bleaching plant from 64.6 to 71 °C (Table 5-1: stream1), 27.8 kg/s hot water in evaporators from 57 to 77 °C (Table 5-1: stream 3) and 130 kg/s of the white water in the pulp machine from 62.1 to 67.1 °C (Table 5-1: stream 2). Since the mass flow of the white water stream is very high, only a portion of that stream enters the AHP and then merges with the other portion to reach the desired temperature for the entire stream (Figure 5-9).

Table 5-1 Streams available as heat sinks above the pinch point

St.	Section	Description	Tin (°C)	Tout (°C)	Flow (kg/s)	Heat load (kW)
1	Bleaching	Hot water	64.6	71	121.7	3260
2	Pulp machine	White water	62.1	63.1	648.1	2770
3	Power plant	Hot water	57	95	27.8	4430
4	Evap. 2	Liquor	98	98	4.80	2600
5	Evap. 2	Liquor	120	128	14.5	2185
6	Evap. 3	Liquor	125	132	14.0	2290

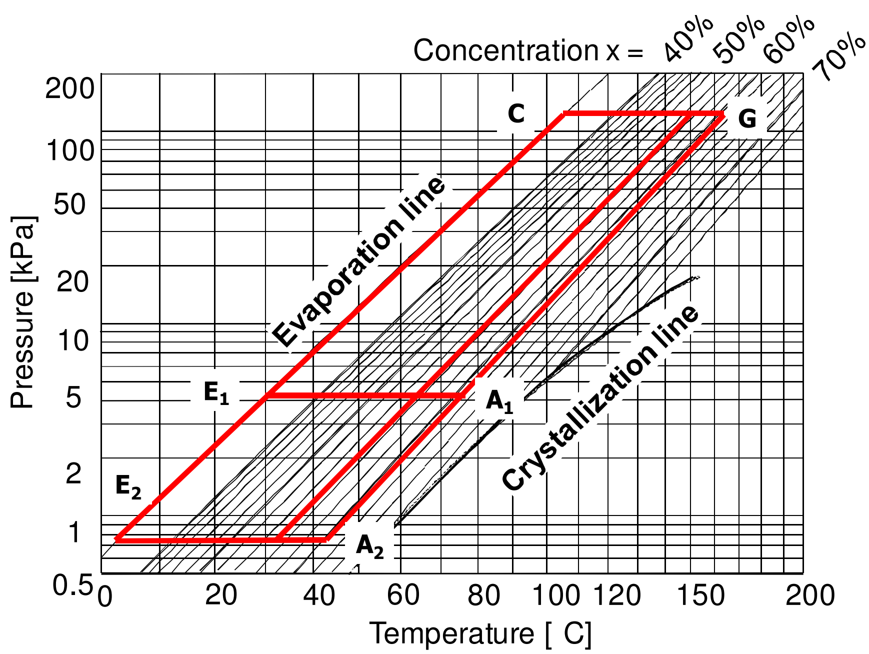

Figure 5-8 The proposed double lift AHP in LiBr/H_2O equilibrium diagram

Table 5-2 Basic design parameters for the proposed double lift AHP

Variable	Unit	Con.	Gen.	Ab.1	Ev.1/Ab.2	Ev.2	SHX1	SHX2
LMTD*	°C	37.18	19.2	8.5	7.9	10.3	19.5	10.8
Pressure	kPa	136.8	136.8	4.1	4.1/0.7	0.7	136.8	4.1
UA**	kW/°C	125	331	430	300	190	200	50
Q	kW	4630	6350	3670	2360	1950	3900	540

* LMTD: Log mean temperature difference
** UA: (Overal heat transfer coefficient)×(Heat transfer area)

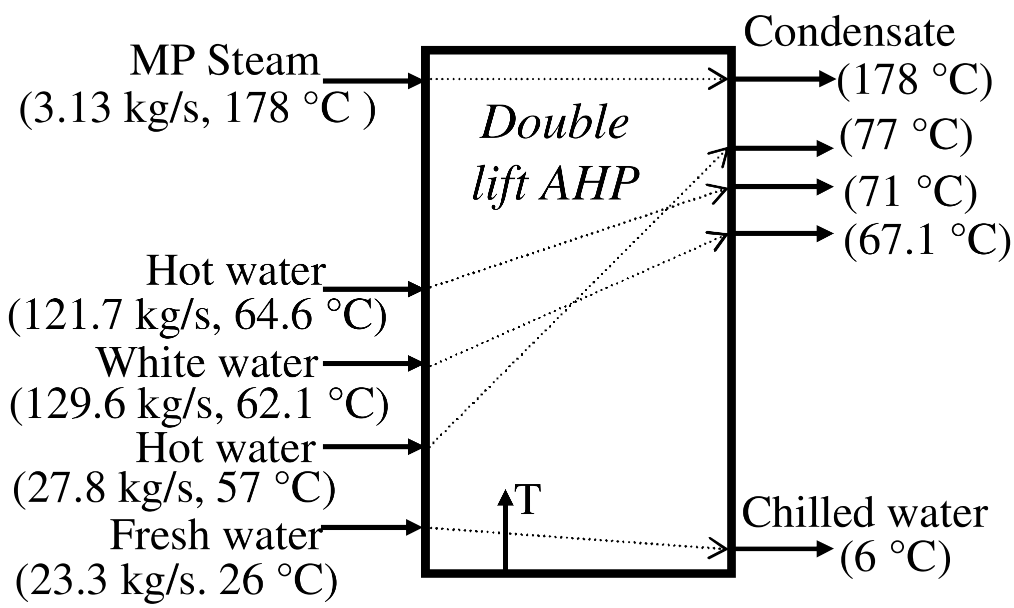

Figure 5-9 Flow diagram of proposed chilled water production in the chemical making plant

The net reduction of steam consumption is 3.6 MW as 2 MW of MP steam used for streams 1, 2 and 3 and 1.6 MW of MP steam used for chilled water production at the current process configuration. Also, the need for 139 kg/s of cooling water and 0.26 MW of electricity is eliminated. The net hot and cold energy demand reductions represent 4 and 17 % of the MHR and the MCR, respectively.

Also, GHG emissions are reduced since less fossil fuel is used to produce steam. The GHG are computed on the basis of specific factors depending on the type of fuel [22]. Implementation of the AHP reduces the GHG emissions by 5300 t/a as CO_2, 130 kg/a as CH_4 and 455 kg/a as NO_x. This aspect of the project would increase the return on the investment if government incentives were made available.

5.8 Opportunity II

The second opportunity is also identified in the pulp bleaching plant. Under current process configuration, bleaching effluents is sent to the sewer. It is a heat source below the pinch point that could be used for an AHP. Table 5-3 shows the properties of the two alkaline and acidic effluents. The heat sinks above the pinch point have a limited load capacity and do not allow for upgrading all the available energy of the alkaline and acidic effluents (potential of 19.4 MW); the alkaline effluent is hence selected as the sole heat source for an AHP. While the heat source load is still relatively high (7.4 MW), a combination of heat sink streams above the pinch point is selected. Stream 4 from Table 5-1 and the hot water in the power plant (stream 3) leaving the last AHP are selected as suitable heat sinks. Unfortunately, streams 5 and 6, which would require a very high temperature lift, can not be used for this case. Considering all the different configurations of AHP/AHT, all AHTs which lead to the very low evaporator temperatures were eliminated from the list of potential candidates. Although double effect AHPs have higher COP than single stage and double lift AHPs, double effect cycles were eliminated since they can not reach the desired temperature lift. Single stage cycles have higher COP and less complexity than double lift cycles, hence the later were eliminated. Single stage AHP using LiBr-H_2O or NH_3-H_2O as the working pair, was the only remaining configuration. The simplified flow diagram of the proposed single stage AHP is presented in Figure 5-10. Since the delivered temperature is high, the working fluid pair best suited for such conditions is LiBr/H_2O. The NH_3/H_2O system would operate at about 50 bar pressure, which is quite a high pressure for such devices. Figure 5-11 shows that the desired temperature lift (40-45 ˚C) can be accomplished in a single effect machine. The absorber and the condenser release their useful heat to increase the temperature of 27.8 kg/s of the hot water in power plant from 77 to 95 °C (Table 5-1: part of stream 3) and to concentrate the liquor (Table 5-1: stream 4). The generator is located at the upper summit of the AHP P vs. T cycle; its operating conditions are: T=152 ˚C, P=94.7 kPa. The best choice for the driving energy in the generator would be MP. Table 5-4 presents the basic design parameters and simulation results for the proposed single stage AHP. A value of 1.68 is obtained for the COP.

Table 5-3 Available heat sources below the pinch point

St.	Section	Description	Tin (°C)	Tout (°C)	Flow (kg/s)	Heat load (kW)
1	Bleaching	Alkaline effl.	56.1	33	77	7430
2	Bleaching	Acidic effl.	55.1	33	130	12000

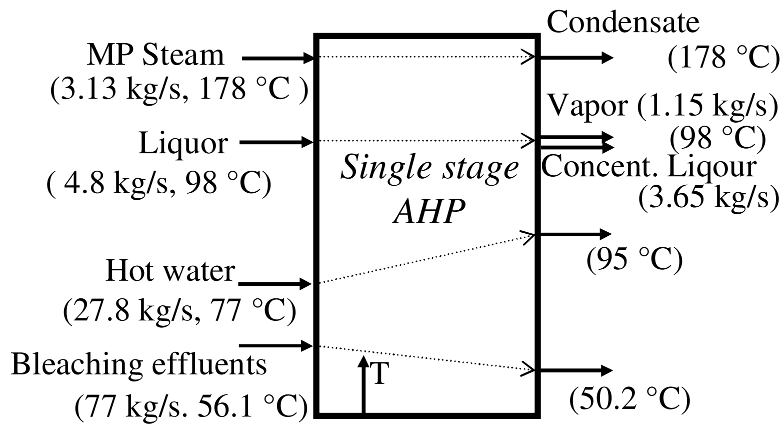

Figure 5-10 Flow diagram of proposed single stage AHP in the bleaching plant

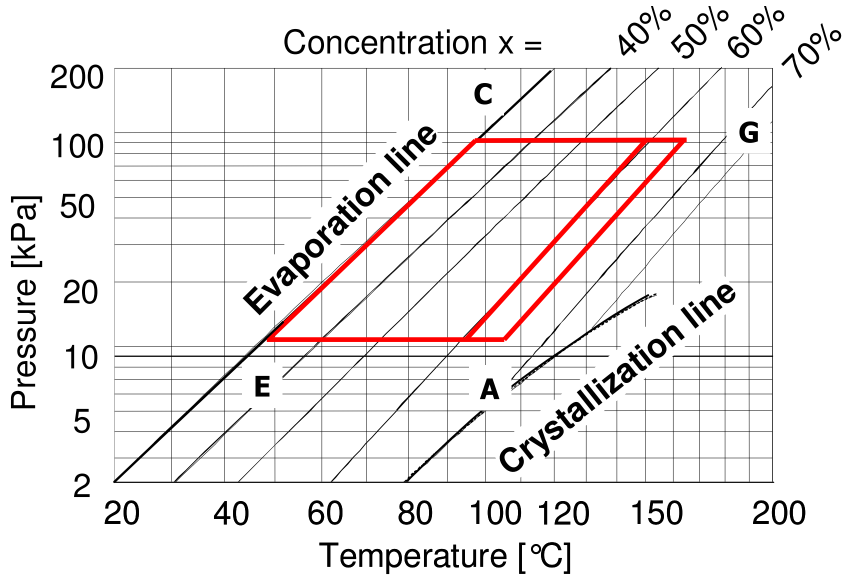

Figure 5-11 The proposed single stage AHP in LiBr/H_2O equilibrium diagram

Table 5-4 Basic design parameters for the proposed single stage AHP

Variable	Unit	Con.	Gen.	Ab.	Ev.	SHX
LMTD	°C	9	12.2	6.3	4.4	26
Pressure	kPa	94.7	94.7	11.3	11.3	94.7
UA	kW/°C	218	230	437	437	33
Q	kW	1962	2806	2753	1923	858

The net reduction of steam consumption is 1.92 MW of MP steam used for stream 3 and 4 (Table 5-1) under the current process configuration. The net hot and cold energy demand reductions represent 2 and 8.6 % of the MHR and MCR, respectively. Based on the same assumptions and data as in opportunity I, the implementation of the proposed AHP reduces GHG emissions by 2560 t/a as CO_2, 65 kg/a as CH_4 and 228 kg/a as NO_x.

5.9 Economic Evaluation

An economic evaluation was performed to determine the economic feasibility of both AHP implementations. The installed cost of each AHP was estimated on the basis of heat cost in $/kW output. For this study, the following costs compiled by US DOE [23] were used; 356 $/kW and

297 $/kW for single stage and double lift AHP, respectively and indexed5; those costs are almost identical to the one presented by Berntsson et al. [24].

The annual operating time of the plant is 8000 hours. It must be noted that production of chilled water is required only during 3400 hours. Electricity and steam costs for this particular plant were taken as 85 and 63 $/MWh, respectively and the cost of cooling water was taken as 0.165 $/t, which includes water screening and effluent treatments. Maintenance and other operating costs were neglected. Consequently, the installed cost of the AHP and the corresponding annual savings of steam, electricity and cooling water could be estimated and presented in Table 5-4. The simple payback time, given by SPB = Investment/Yearly Savings, was also calculated. An increased operating time for the first case to meet additional chilled water needs would further reduce the pay back time.

Table 5-5 Cost analysis for the two cases

Case	Annual savings				Installed cost	SPB
	Electricity (MW)	Steam (MW)	Cooling water (t/s)	Total (M$/a)	(M$)	(a)
Opp. I (Double lift)	0.26	3.57	138.9	0.92	2.46	2.7
Opp. II (Single stage)	0	1.92	0	0.97	1.61	1.7

Life-cycle cost analysis of the two cases was performed using the net present value method (NPV) [9]. The initial investment cost and the yearly energy savings were estimated over the life cycle. A common discount rate of 7% and a yearly escalation rate of 4% for fuel price have been used for the time span of the investment, which was set at 15 years. Figure 5-12 shows the

[5] All costs in this paper are given in 2008 Canadian dollars

evolution of the NPV over 15 years; the abscissa gives the time elapsed from the date of investment and the intersection with the nil value line indicates the time required for the projects to become cost-effective. For these cases it is around 3 and 1.9 years for the double lift and single stage AHPs, respectively. The figure shows that the installation of the AHPs would be profitable not only in the short term (quite short SPB), but also in the long term. It should be noted that neither operating costs nor potential GHG emission credits were taken into account. The latter could be a significant factor of payback time reduction.

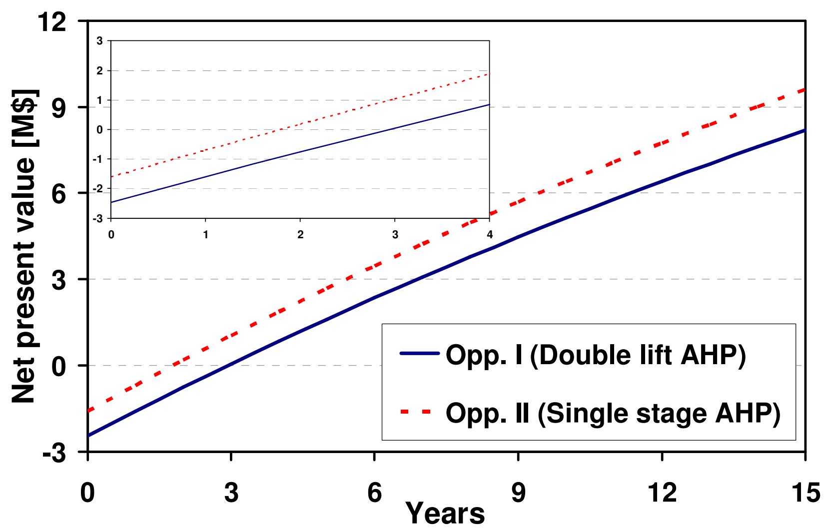

Figure 5-12 Net present value for the two case studies

5.10 Conclusion

Two generic opportunities for the implementation of AHPs in a Kraft pulping process have been identified and described. They emphasize the following points:

- It is critical to position the absorption heat pumps in the process on the basis of a reliable methodology considering Pinch Analysis, process and AHP constraints to ensure a real and feasible net energy gain.

- Water closure and condensate return should be performed before heat exchanger network optimization and implementation of AHPs.
- The heat exchanger network of the process should be reconfigured to yield improved conditions for AHP implementation, hence enabling further decrease of heating and cooling requirements of the process. This reconfiguration only marginally increased the cost of the heat exchangers required.
- Simple pay back times of 2.7 and 1.7 years have been estimated for the two presented cases. The implementation of case II is more attractive as it produces a shorter SPB and less installation cost, while generating almost the same yearly saving.
- Although the identified opportunities were presented for a particular Kraft mill, the idea of implementing an AHP in the bleaching plant to produce chilled water and the use of wasted energy in the effluents could be extended to all Kraft mills and therefore became generic implementations. Specific heat sinks above the pinch point and AHP designs might vary from one plant to another, but one should always be able to identify and optimize them based on the presented methodology.

The economics of AHPs remain uncertain in the current environment that the Canadian P&P industry is facing. However, those energy upgrading devices have strong advantages in a sustainable development perspective, and GHG abatement credits could make them more economically attractive.

5.11 Acknowledgments

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CHAPTER 6 - A MODEL FOR ANALYSIS AND DESIGN OF H_2O-LIBR ABSORPTION HEAT PUMPS

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Keywords: Absorption heat pump, H_2O-LiBr, modeling, experimental results.

6.1 Presentation of the Article

In this article, a single-stage H_2O-LiBr absorption heat pump is experimentally characterized and modeled. A design and dimensioning model of H_2O-LiBr absorption heat pumps was developed. Close agreement between experimental and simulation results was found. The capability of the model is illustrated by dimensioning an absorption heat pump implemented in a Kraft process. This article is submitted to Energy Conversion and Management Journal.

6.2 Abstract

An experimental and simulation analysis of a laboratory single-stage H_2O-LiBr absorption heat pump with a cooling capacity of 14 kW has been performed. Design characteristics of the machine are given and experimental results obtained from the variation of the five most influential parameters are presented. The machine performance, as described by the coefficient of performance (COP) and cooling capacity was then measured at different flow rates and temperatures of the external cool and hot water loops and for different temperatures of produced

chilled water. A design and dimensioning model of H_2O-LiBr absorption heat pumps was developed. First, the steady-state simulation results of the model were compared with experimental measurements. Close agreement between experimental and simulation results was found. Results also show that the heat pump can adequately operate over a wide range of generator input energy and chilled water temperature; the cooling water flow rate and temperature significantly affect the performance of the machine. Finally, the capability of the model is illustrated by dimensioning an absorption heat pump implemented in a Kraft process.

6.3 Nomenclature

Nomenclature

A	surface area (m^2)
D	tube diameter (m)
F	fouling factor (m^2°C /kW)
k	thermal conductivity (kW/m^2°C)
h	enthalpy (kJ/kg)
LMTD	log mean temperature difference (°C)
m	mass flow (L/min)
P	pressure (bar)
Q	heat capacity (kW)
T	temperature (°C)
U	overall heat transfer coefficient (kW/m^2-°C)
un	uncertainty
x	concentration of LiBr (%)

Greek symbols

α	heat transfer coefficient (kW/ m^2-°C)
Γ	fluid mass flow rate per unit length per one side of the exchanger tube (kg/s-m)

Abbreviations

Ab	absorber
Act	actual
AHP	absorption heat pump
Con	condenser
COP	coefficient of performance
Eva	evaporator
Exp	experimental
Gen	generator
RTD	resistance temperature detector
SHX	solution heat exchanger
Sim	simulation
Theo	theoretical

Subscripts

a	absorber
c	condenser
e	evaporator
eq	equilibrium
g	generator
in	inlet
out	outlet
ref	refrigerant
sat	saturation
sol	solution
st	strong solution
w	water
we	weak solution

6.4 Introduction

Absorption Heat Pumps (AHP) have been used since the late 19th century and a large body of scientific and technical literature has been devoted to the fundamental principles, engineering design and application of those devices [1-3]. Due to volatile energy prices and environmental concerns, these systems have received more attention in the past two decades. AHPs are attractive if they are implemented correctly and supplied with waste energy or water heated through solar collectors. Such devices are environmentally friendly since they use working fluids which do not cause ozone depletion. For the majority of AHPs used in industrial applications, H_2O-LiBr is the working fluid pair of choice since it is not toxic, has a high enthalpy of vaporization and does not require a rectification step.

AHP chillers working at low driving temperatures are suitable for trigeneration (cold, heat, and power production simultaneously) and solar cooling applications. They commonly have been used in air conditioning and it has been shown that they can be effectively integrated to load shifting during summer peak electricity demand. However, they are rarely commercially available with a small cooling capacity.

Single effect AHPs have been investigated in the past [4-7]. Many parameters can affect the performance of AHP systems and several studies have been reported in the literature [8-11]. Mathematical models of various complexity have been developed for different purposes, such as single simulation of absorption system, evaluation of potential working fluids, optimization and design [5, 6, 10, 12, 13]. Some of those models were validated against experimental results with good agreement. However, most of them are restricted to a specific system and are empirically derived and in some cases overly simplified.

In order to characterize the performance of the AHP and give an experimental dimension, a prototype of 33 kW AHP was built through an international collaboration [14-16]. In a previous study, to determine limits of operation of this prototype and to characterize the performance of the machine, 16 selected experiments based on a partial fractional design with seven variables was performed [14]; those variables were the temperature and flow rate of chilled, cooling and hot water as well as the internal solution mass flow rate. The respective effects of the independent variables on the COP and the heat fluxes in the main four heat exchangers were computed. The variations of COP were very large (from 0.1 to 0.75); while some results were close to the design COP of 0.75, others were out of bound. The temperature of chilled, cooling and hot water and, the flow rate of cooling water and hot water were found to be the most influential operating parameters.

In the present study, those preliminary results were used to design an experimental plan aimed at measuring the performance of the machine under different operating conditions using the five previously identified dominant parameters. In parallel, a design and dimensioning model of the absorption cycle was developed. Experimental results were then used to validate the model. This model requires minimal information about the working fluid and characteristics of the heat exchangers, and it is suitable for rapid and reliable simulation of an existing absorption cycle, evaluation and comparison of different working fluids and design and dimensioning of new AHPs. Finally, the model is used for dimensioning an AHP implemented in an actual Kraft process.

6.5 Experimental Setup

6.5.1 Absorption Machine

The prototype under study is a single effect AHP unit illustrated in Figure 6-1 [15]. It consists of four main heat exchangers: condenser, generator, absorber and evaporator and a fifth auxiliary heat exchanger placed between the generator and the absorber, the solution heat exchanger SHX. The four main exchangers are of shell-and-tube type with horizontal tube bundles and a pressure-loss free arrangement for the distribution of the working solution and refrigerant. The solution heat exchanger is a commercially available plate heat exchanger. The working pair is H_2O-LiBr, in which water is the refrigerant and an aqueous lithium bromide solution is the absorbent. In addition to the main working fluid, a surfactant additive, Octyl alcohol (2-ethyl-1-hexanol), is used to act as a heat transfer enhancer on the solution side. The prototype is designed to offer good controllability with a high coefficient of performance (COP), in a compact design. To minimize pressure drops and prevent leakage problems, condenser and generator, as well as evaporator and absorber are placed together in a single vessel. The machine was designed for cooling purposes, either for the production of chilled water for air-conditioning systems with an evaporator temperature ($T_{e,out}$) of 6-12 °C, or for the supply of cooling to radiant ceilings with an evaporator temperature of 15-18 °C.

Table 6-1 presents the design specification of the absorption machine. The absorber and condenser are water-cooled. The cooling water is heated in nominal conditions from 27 to 40.4 °C ($T_{a,in}$ to $T_{c,out}$). It is connected in series: first the absorber is cooled and then the condenser. Series arrangements of the condenser and absorber allows for easy operation since a single pump can be used without the control problem associated with a parallel design. It is also a better configuration to avoid crystallization in the system. During the experiments, the temperature level of the outlet cooling water ($T_{c,out}$) could be raised to 44 °C. This temperature is sufficient for a heat pump supplying a low-temperature heating systems such as those used for floor heating. The hot water ($T_{g,in}$) is supplied at 95 °C, making the machine suitable for use in cogeneration units driven by engines or fuel cells, or in solar-assisted heating and air-conditioning systems. A centrifugal pump is used for the water recirculation in the evaporator. The solution pump is a gear pump that operates with a magnetic drive. The advantage of this type of pump is the seal free drive that provides a hermetically enclosed environment without emission. Flow rates of recirculated water and solution circuit are controlled by diaphragm valves bypassing the pumps. Ball valves just before and after the pumps were built in for maintenance. Both valves can be closed and the pump can be replaced without losing solution or water. There is a bypass between the evaporator recirculation pipe and the solution circuit just before the solution enters the generator. Normally this bypass is closed and is used only in case of emergency to dilute the solution in the solution loop.

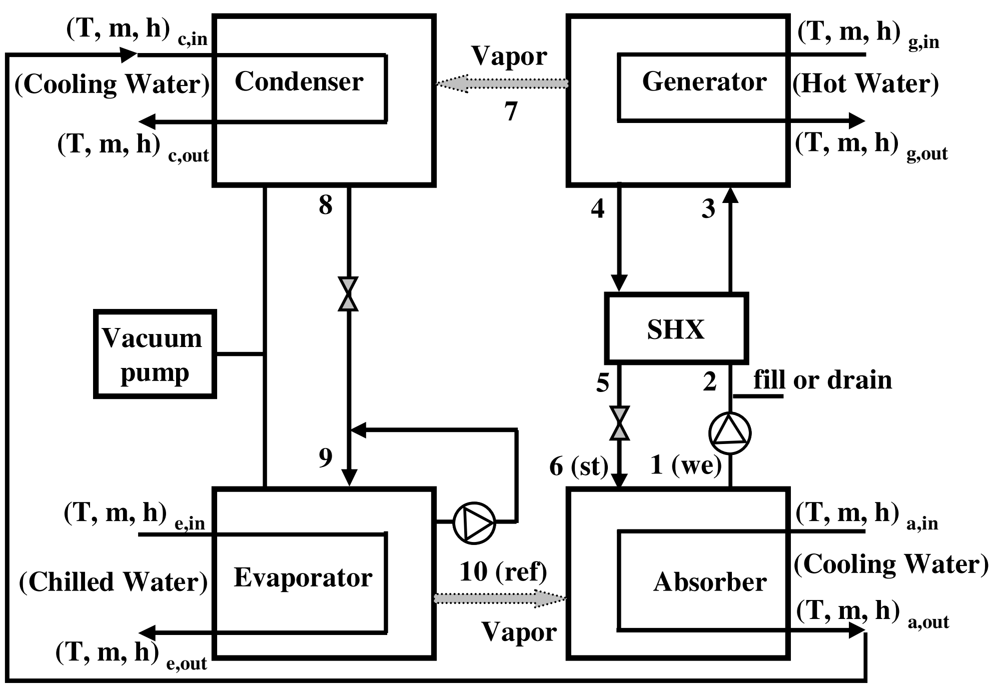

Figure 6-1 Schematic of the absorption heat pump

Table 6-1 Design specification of the absorption machine

Working Fluid	H_2O-LiBr
Cooling capacity (Q_e)	14 kW
Chilled water temperature in/out ($T_{e,in}/T_{e,out}$)	21.7/15 °C
Flow rate ($m_{e,in}$)	30 L/min
Driving heat capacity (Q_g)	18.6 kW
Hot water temperature in/out ($T_{g,in}/T_{g,out}$)	95/81.7 °C
Flow rate ($m_{g,in}$)	20 L/min
Rejected heat capacity ($Q_{a+}Q_c$)	32.6 kW
Cooling water temperature in/out ($T_{a,in}/T_{c,out}$)	27/40.4 °C
Flow rate ($m_{a,in}$)	35 L/min
COP	0.75

6.5.2 Power Station

The power station provides constant conditions at the desired temperatures and flow rates in the external circuits. It consists of three independent water loops that are connected to the heat pump. Figure 6-2 shows these external loops providing the heat sources and sinks required by the prototype. The hot water loop is connected to the generator and supplies the driving energy. Three heaters of 10 kW heating capacity each, heat the water to a set–point temperature. The power of the heaters is controlled by the generator inlet temperature. The cooling water loop cools the absorber and condenser. It is connected in series: the absorber is cooled first and then the condenser. Normally the loop is connected to an external cooling tower. For the purpose of these experiments, the cooling tower is simulated by a plate heat exchanger connected to the water network of the laboratory. Water from the network is constantly running through the heat exchanger. Its flow rate is controlled by a manual valve. A three-way valve on the side of the cooling water loop is used to control the absorber inlet temperature. The chilled water loop simulates the user demand for cooling. The user could be, for example, a house that needs cooling to compensate for loads generated by occupants and by solar radiation. These loads are simulated by two heaters of 10 kW heating capacity each. The working principle is the same as for the hot water loop. Three centrifugal pumps are used for water recirculation in the power station. The flow is controlled by a control gate valve placed at the discharge of each pump. The flow rates of each water loop can be adjusted by slider valves.

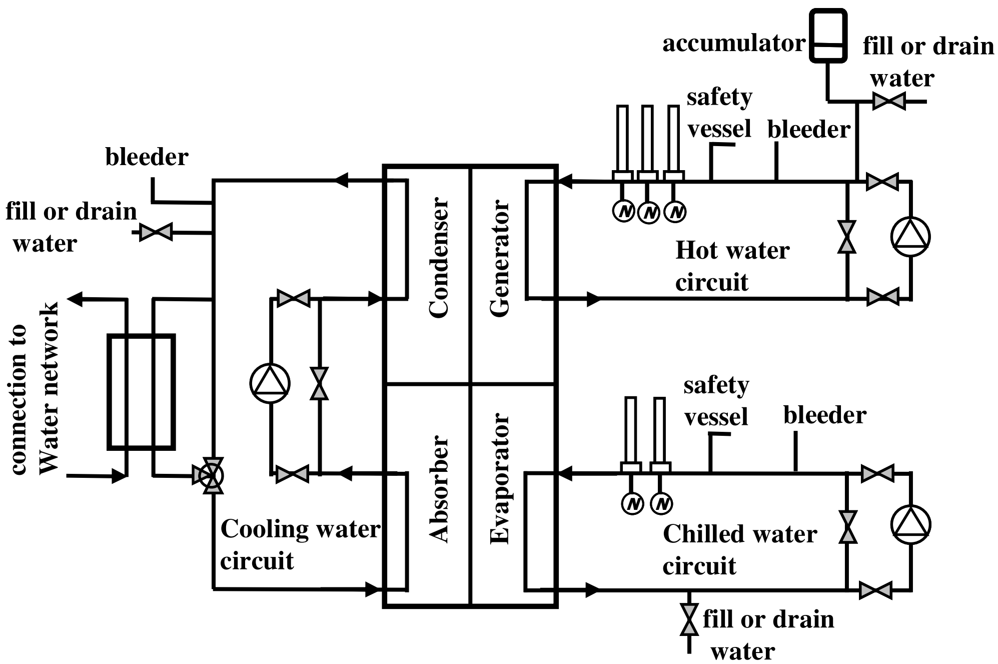

Figure 6-2 Schematic of the power station system

6.5.3 Measurement equipment and data acquisition

The prototype is computer-monitored and -controlled by means of a Labview program [17]. It is equipped with 23 sensors to record all process pressures, temperatures and flow rates. On the cooling machine itself, there are 8 resistance temperature detectors (RTD) to measure 8 internal stream temperatures. Two flow meters measure the volume flow rates of the recirculation loop at the evaporator and in the solution circuit. The flow meter in the solution circuit can also measure the density of the solution rich in water (weak solution). Two pressure sensors measure the pressure in the evaporator and in the condenser. Furthermore, there are three sight glasses in the circuit to provide visual observation during the experiments and one sight glass on each of the four main heat exchangers. There are 8 additional RTD temperature detectors in the power station to measure the 8 external stream temperatures at the inlet and outlet of each main heat exchanger. Three flow meters are installed to measure the volume flow rates of the three external water loops. Ten manometers in the external loops indicate the water pressure during the experiments, but the data is not registered. These modules receive and send signals from and to the measuring instruments and communicate with the data acquisition program. This program controls the pumps and heaters and saves the experimental data at selected time intervals.

Real time operating variables and certain calculated parameters are constantly displayed on the monitor during the experiments. The calculated parameters are saved together with the measured experimental data in a worksheet. Calculated parameters are the loads of the four main heat exchangers, the COP, the concentration of the weak and strong solutions and the solution recirculation flow rate. The COP for cooling and heating are calculated as follows:

$$COP_{cooling} = \frac{Q_e}{Q_g} \tag{1}$$

and

$$COP_{heating} = \frac{Q_a + Q_c}{Q_g} \tag{2}$$

6.6 Model Development

An analytical model to simulate a single stage AHP with H_2O-LiBr was developed; its purpose is to assist in the design and dimensioning of those thermal machines. In this model, each component is treated as a control volume with its own inputs and outputs. The performance of the cycle is described by mass balances on water and LiBr and energy balances for each machine element and, by overall energy balance and heat transfer equations between the internal and external streams. Solutions in the generator and absorber are considered to be in equilibrium with refrigerant vapor at the same temperature and pressure. Major assumptions made to simplify the simulation and analysis are: steady-state operation, negligible heat losses and heat gains between the system and its environment, negligible work input to all internal pumps as compared to input energy to the generator and negligible pressure losses in the equipment and pipes. The empirical equations of enthalpies, temperatures, concentrations and vapour pressures of the H_2O-LiBr working pair given by McNeely are used [18]. The IAPWS-IF 97 equations has been used for calculating the thermodynamic properties of water and steam [19]. The boiling temperature of saturated aqueous lithium bromide solution is calculated as a function of the saturated temperature of the vapor and concentration of the solution proposed by Feuerecker et al. [20].

The heat transfer rate between the internal and external streams in the generator is given by,

$$Q_g = U_g A_g LMTD_g \qquad (3)$$

where A_g is the total heat transfer surface area in the generator, U_g is the overall heat transfer coefficient for the generator and $LMTD_g$ is the logarithmic mean temperature difference defined as:

$$LMTD_g = \frac{(T_{g,in} - T_4) - (T_{g,out} - T_7)}{\ln \dfrac{T_{g,in} - T_4}{T_{g,out} - T_7}} \qquad (4)$$

Prediction of heat transfer coefficients is affected by many uncertainties and unexpected results often arise in practice [7]. For the purpose of this simulation, the user can choose between average overall heat transfer coefficients for each heat exchanger calculated from experimental results if they are available, and heat transfer coefficients calculated from correlations. The overall heat transfer coefficient for the generator is determined from the sum of heat transfer resistances based on the outside surface of the tube as:

$$U_g = \frac{1}{[\frac{D_{out}}{D_{in}}][\frac{1}{\alpha_{g,in}}] + [\frac{D_{out}}{2k}]\ln[\frac{D_{out}}{D_{in}}] + \frac{1}{\alpha_{g,out}} + [\frac{D_{out}}{D_{in}}]F_{in} + F_{out}} \qquad (5)$$

For this equation, the value of the fouling factors (F_i and F_o) for the outside and inside surfaces of the tube can be assumed as 0.09 m^2°C /kW [21]. A correlation for copper thermal conductivity, k, was developed by integrating table information provided by Holman [22], resulting in equation (6):

$$k = -3e - 7T^3 + 0.0002T^2 - 0.0832T + 385.99 \qquad (6)$$

The internal heat transfer coefficient between hot water and the generator tube surface ($\alpha_{g,in}$) is determined using the Gnielinski correlation [23]. If steam is used as the driving energy for the generator, the correlation presented by Agrawal for the condensation of vapours flowing inside a cylindrical tube is used [24].

There are few available published data for the generator external heat transfer coefficient ($\alpha_{g,out}$). A new correlation for $\alpha_{g,out}$ was produced for the purpose of this work by integrating experimental results published by Wang et al. [25], resulting in Equation 7:

$$\alpha_{g,out} = 5554.3\Gamma^{0.236} \quad (7)$$

The definition of Γ used here is the total mass flow rate of the internal fluid (LiBr solution in this case) per unit length of wetted tube (only one side of each tube is wetted).

Equations analogous to equations 3, 4 and 5 are applied to the absorber, condenser and evaporator. The Gnielinski correlation [23] is used for all internal heat transfer coefficients. The external heat transfer coefficient for the condenser is computed by a correlation for laminar evaporating films proposed by Holman [22]; in the case of the evaporator the corresponding correlation developed by Rohsenow et al. [26]was used. The external heat transfer coefficient for the absorber is based on the correlation presented by Hoffmann et al. [27].

The SHX is characterized by using a solution heat exchanger effectiveness (ε_{SHX}) defined by equation 8.

$$\varepsilon_{SHX} = \frac{T_4 - T_5}{T_4 - T_2} \tag{8}$$

The entire set of non-linear equations is solved with the Gauss-Newton method.

The model has two different fields of applications. It can be used either to analyze the performance of an existing machine under various operating conditions or to design and dimension a new one. The organigram of the model in the simulation and analysis mode is shown in Figure 6-3. Results from the simulation were validated by experimental results generated on the laboratory prototype and by results from the literature [1, 7]. Mass and energy balance equations for a single stage AHP used in the model are given in Appendix A.

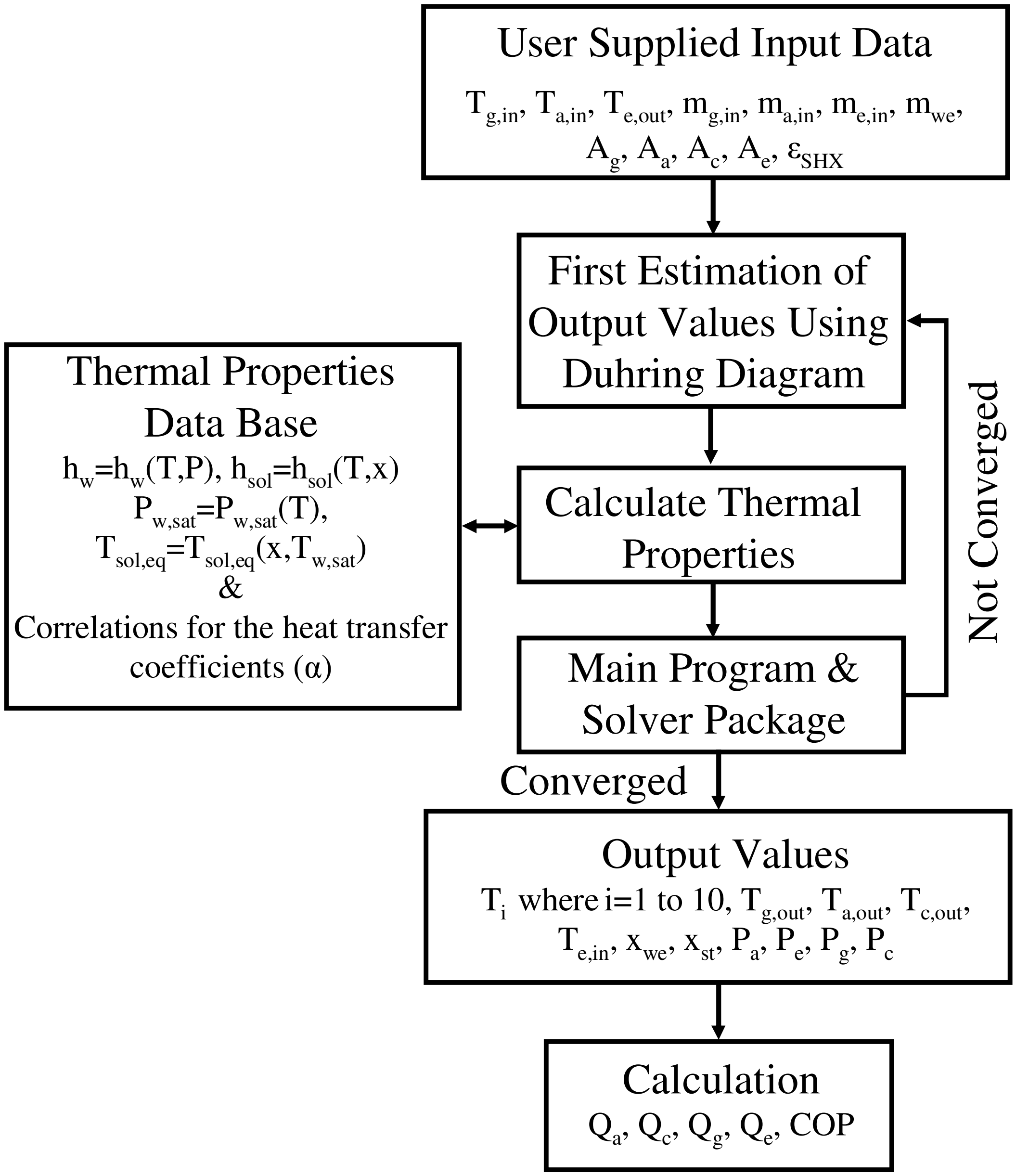

Figure 6-3 Block diagram of the simulation program

The core of the model is unchanged when it is used for design and dimensioning purposes. However, user input data and output values are different. The user must, in that case, define external temperatures and mass flow rates, concentration of the strong solution (x_{st}) and solution heat exchanger effectiveness (ε_{SHX}) as input data. The model calculates all the internal temperatures, pressures and mass flow rates as well as heat transfer coefficients for the heat exchangers as output values. The heat exchangers surface areas are also calculated as design values. The usefulness of the model is presented in the following sections, first, as a tool to predict the operating parameters of an existing AHP and second, as a tool for design and dimensioning of an AHP for a particular industrial application.

6.7 Model Validation and Experimental Work

6.7.1 Experimental Plan

Five sets of experiments were conducted to measure the performance of the machine under different operating conditions. It focused on the effect of variations in the five dominant variables as identified in the study presented earlier [14] on the performance of the heat pump. Each of those variables was varied over a given interval in a stepwise fashion by a predetermined increment, other variables remaining constant at their median value. The results by Jahnke et al. [14] provided guidance to chose the modalities so that each variable would span the practical range of operation for a single effect AHP (Table 6-2). After each temperature or flow rate change, steady state was considered as being achieved when all recorded variables remained constant for about 20 minutes. A method for estimating the uncertainty on experimental results by Kline and McClinton [28] was used; it is summarized in Appendix B.

Table 6-2 Modalities for the input variables

	$T_{e,out}$ (°C)	$T_{a,in}$ (°C)	$T_{g,in}$ (°C)	$m_{a,in}$ (L/min)	$m_{g,in}$ (L/min)
Min	6	20	65	15	10
Nominal	15	27	95	35	20
Max	16	40	110	43	35
Increment	2	4	5	5	5

6.7.2 Results

Figure 6-4 shows the variation of the COP and cooling capacity of the absorption heat pump as a function of the temperature of the heat source to the generator ($T_{g,in}$). The uncertainty bar for the experimental data is also displayed in this figure, considering (1% T) and (2% m) as the uncertainties for the temperature sensors and flow meters, respectively. Computed uncertainties are 10.3% for the COP and 2.9% for the cooling capacity in the experiments. The range of relative uncertainty being predictably constant in all experiments, it is not presented in the following figures. The experimental COP is almost constant close to the design COP with a maximum value of 0.76 at $T_{g,in}$ = 90 °C. The capacity varies almost linearly starting from a low value of 5.2 kW up to 16.5 kW. The fact that the capacity is increasing while the COP remains constant indicates that there are several factors with opposite effects that vary as the heat source temperature changes.

The performance of the heat pump is strongly affected by changes in the internal and external temperatures. The COP would be expected to increase with increasing generator temperature. At the same time, the evaporator temperature is increased which results in a decrease of the COP. This negative effect seems to balance the beneficial effect of the high temperature of the generator inlet. Two pairs of simulated COP and cooling capacity results are also presented in Figure 6-4. In pair A, the model uses average overall heat transfer coefficients calculated from experimental results. In pair B, the model calculates heat transfer coefficients using the correlations presented earlier. In general, there is a good agreement between measured and calculated COP and cooling capacity by both methods. Since similar behavior was observed in the other experiments; simulation results based on values of U computed from the correlations are not presented in the following figures.

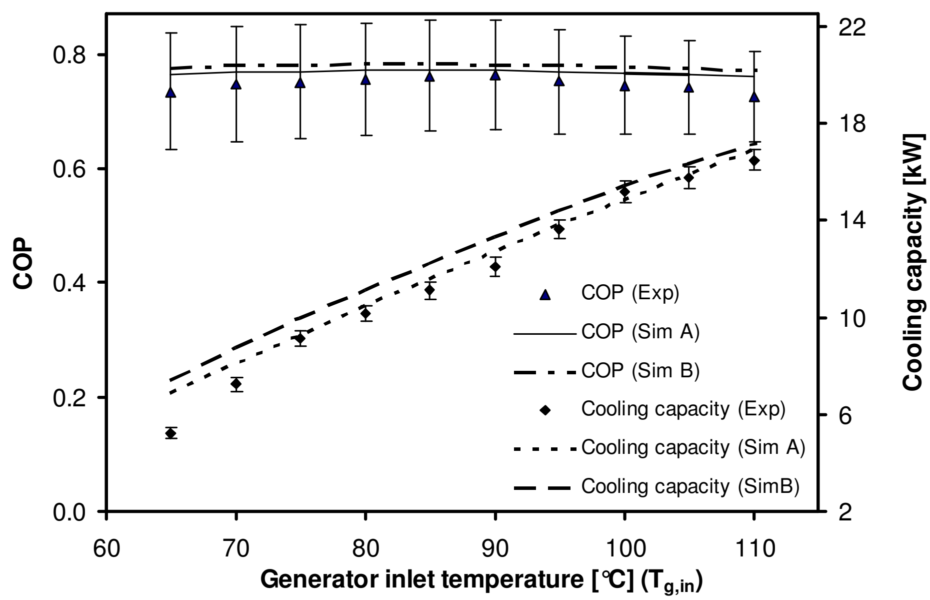

Figure 6-4 Variation of the COP and cooling capacity with generator inlet temperature ($T_{g,in}$)

Table 6-3 presents experimental data and model predictions for a set of experiments in which the effect of generator inlet temperature on the performance of the AHP is studied. It also shows the good agreement between measured and predicted values. Mass flow rates for the three external loops were those in Table 6-1 i.e. design values.

Table 6-3 Effect of generator inlet temperature ($T_{g,in}$) on the performance of AHP

Exp/Sim	Independent variables (Input)			Process variables (Output)				Calculated values				
	Temperature (°C)							Heat load (kW)				COP
	$T_{g,in}$	$T_{a,in}$	$T_{e,out}$	$T_{g,out}$	$T_{e,in}$	$T_{c,out}$	$T_{a,out}/T_{c,in}$	Gen	Ab	Eva	Con	
Exp 1	70.1	26.9	14.8	63.2	18.3	33.9	30.2	9.7	8.1	7.3	8.9	0.75
Sim 1	70.1	26.9	14.8	62.5	18.7	34.7	30.6	10.6	8.9	8.1	9.8	0.77
Exp 2	79.9	26.8	14.9	70.2	19.8	36.6	31.6	13.5	11.6	10.2	12.1	0.76
Sim 2	79.9	26.8	14.9	70.2	19.9	36.8	31.6	13.5	11.8	10.5	12.2	0.77
Exp 3	89.9	27.1	15.2	78.5	21.0	38.7	32.8	15.9	13.8	12.1	14.2	0.76
Sim 3	89.9	27.1	15.2	78.1	21.3	39.0	33.0	16.5	14.4	12.7	14.8	0.77
Exp 4	100.1	27.2	15.1	85.4	22.4	41.9	35.0	20.4	18.9	15.2	16.7	0.74
Sim 4	100.1	27.2	15.1	86.2	22.2	41.1	34.6	19.4	18.2	14.9	16.1	0.77
Exp 5	109.7	26.8	14.9	93.4	22.8	43.1	35.7	22.8	21.5	16.5	17.8	0.73
Sim 5	109.7	26.8	14.9	93.7	23.0	43.1	35.6	22.3	21.4	16.9	17.8	0.76

Values of overall heat transfer coefficients calculated from experimental results, designated "Experimental" and those computed from a correlation designated "Correlations" for the different heat exchangers are presented in Table 6-4. As expected in general the experimentally computed ones are lower than the ones computed from correlations. The biggest difference between the two sets is in the case of the generator where the external heat transfer coefficient is calculated using the new correlation (Equation 7). This can be explained by the fact that the operating range of this particular heat exchanger was outside the range of literature data [25] used to obtain this correlation, thus requiring an extrapolation. A good agreement between calculated COP by both methods (Figure 6-4) was observed which shows that the COP is relatively insensitive to change in U while the capacity is more sensitive to those coefficients. It should be noted that the same behavior was observed by Herold et al. [1].

Table 6-4 Overall heat transfer coefficients of the various heat exchangers

Unit	Overall Heat transfer coefficient (kW/m^2K)	
	Correlations	Experimental
Gen	1-1.15	0.7
Ab	0.4-0.65	0.52
Eva	1.62-1.65	1.11
Con	2.3-2.4	1.75
SHX	≈ 0.2	0.2

The variation of the COP and cooling capacity with chilled water temperature ($T_{e,out}$) is shown in Figure 6-5. The COP varies very little over the experimental range of chilled water temperature (6 °C to 16 °C). This variation has a stronger effect on the cooling capacity which increases from 9.2 kW at 6 °C to 14.3 kW at 16 °C. It causes temperature effects in the system that balance the increased capacity and the COP remains almost constant. The COPs predicted by the model and measured are almost identical. The difference between measured and predicted cooling capacities at lower chilled water temperature can be explained by the fact that average values of overall heat transfer coefficients for each heat exchangers are used for the simulation. Those average valued may differ from the actual heat transfer coefficients in the case of low generator inlet temperatures.

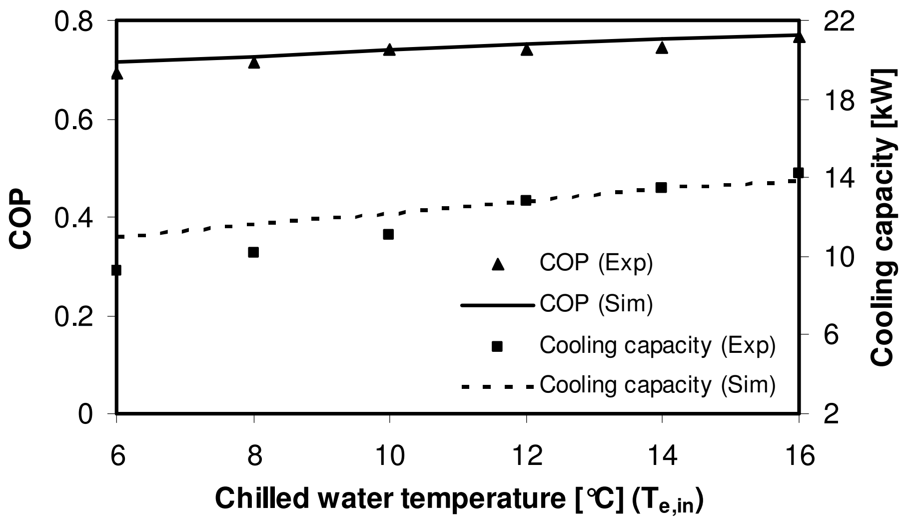

Figure 6-5 Variation of the COP and cooling capacity with chilled water temperature ($T_{e,out}$)

As shown in Figure 6-6, cooling water temperature ($T_{a,in}$) has the strongest effect on the cooling capacity of the heat pump. There is good agreement between experiments and simulations. The cooling capacity increases significantly as cooling water temperature decreases. In an actual implementation, low cooling water temperature can be achieved using a wet cooling tower, while higher cooling water temperature can be obtained with dry cooling towers. Once again, the COP varies only slightly in the range of 0.78 to 0.7 with cooling water temperature between 20 °C and 32 °C as the combined effects of temperature and capacity changes tend to cancel each other. At higher cooling water temperatures, the COP drops to 0.53. This is due to the fact that the rate of decrease of the evaporator load becomes larger than that of the other heat exchangers at the higher cooling water temperature.

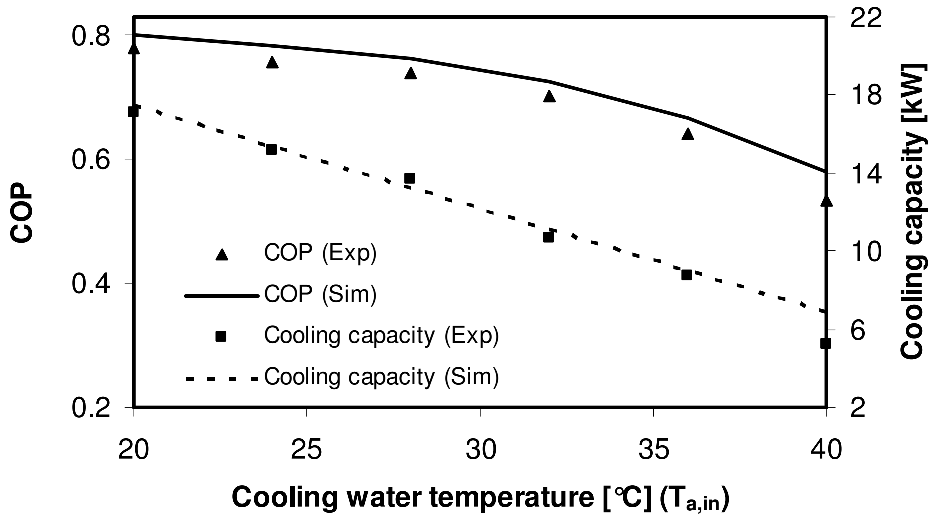

Figure 6-6 Variation of the COP and cooling capacity with cooling water temperature ($T_{a,in}$)

Figure 6-7 shows that the generator water flow rate ($m_{g,in}$) has little effect on the performance of the heat pump. The COP is almost constant in the range of flow rates which indicates that the hot water side is not the dominant factor in controlling heat transfer coefficient of the generator. The cooling capacity is improved slightly by high generator flow rate. The effect of cooling water flow rate ($m_{a,in}$) on the cycle performance is plotted in Figure 6-8. The cooling capacity and the COP are improved by high cooling water flow rates. It indicates that the cooling water side is the dominant factor in the heat transfer coefficients of the absorber and condenser.

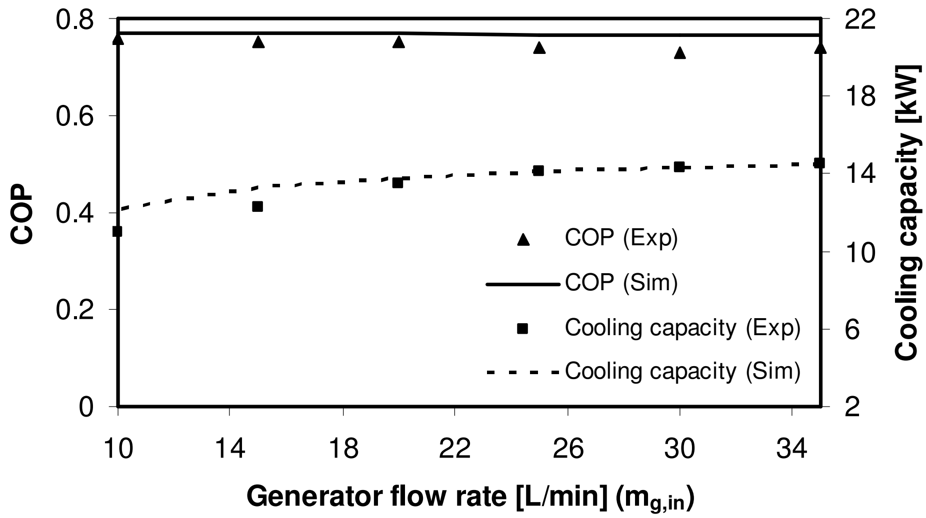

Figure 6-7 Variation of the COP and cooling capacity with generator flow rate ($m_{g,in}$)

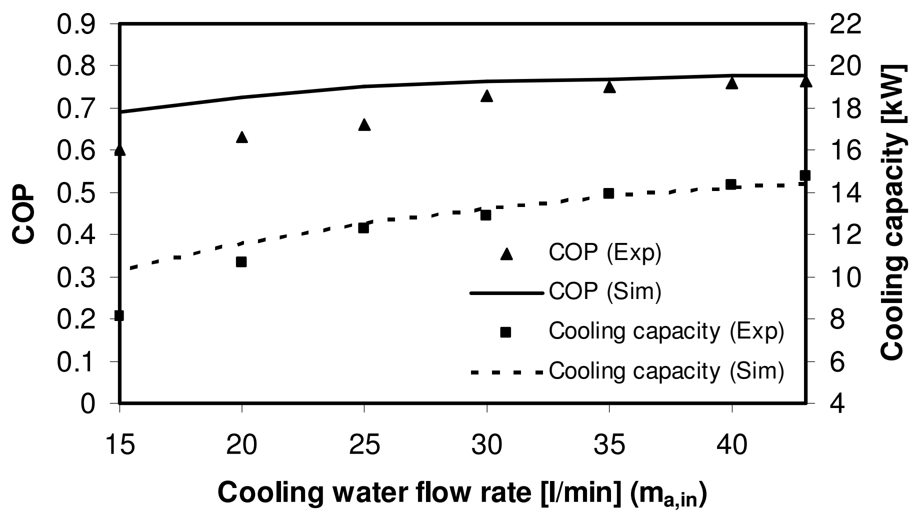

Figure 6-8 Variation of the COP and cooling capacity with cooling water flow rate ($m_{a,in}$)

It is worth mentioning that, in this study higher cooling capacities and better COPs are obtained, in comparison with the results presented by Jahnke et al. [14]. Table 6-5 presents selected experimental results (4 runs among 16) presented in [14]. The higher performance observed in our study can be explained by the different experimental conditions of the two studies. In [14], the AHP was operated at conditions close its operating limits with more than one variable being kept at its extremum. In our study, the performance of the heat pump is observed with only one variable being varied over a given interval, all other variables being kept constant at their design value. In comparison with the design performance, experiment J1 from Table 6-5 has the same COP but lower cooling capacity. It can be explained by its low generator inlet temperature ($T_{g,in}$); as discussed in the previous sections. On the other hand, experiment J4 has the lowest performance of those presented. It is because of combining the negative effects of high cooling water temperature ($T_{a,in}$), low generator inlet temperature ($T_{g,in}$) and very low cooling water and generator flow rates ($m_{a,in}$ & $m_{g,in}$) at the same time.

Table 6-5 Selected experimental results from Jahnke et al. [14]

Exp	Independent variables (Input)						Calculated values				
	Temperature (°C)			Flow rate (L/min)			Heat load (kW)				COP
	$T_{g,in}$	$T_{a,in}$	$T_{e,out}$	$m_{g,in}$	$m_{a,in}$	$m_{e,out}$	Gen	Ab	Eva	Con	
J1	75	26.8	14.8	37.9	40.3	44.7	13.3	12.0	10.0	10.7	0.75
J2	110	26.4	15	4.8	39.2	15.0	14.1	12.9	9.8	10.7	0.70
J3	110	26.5	15	4.9	10.9	44.5	8.2	7.1	4.1	4.9	0.50
J4	75.4	34	25	6.9	13.2	43.5	3.6	2.7	0.3	0.9	0.09

6.8 Application to Design

The model will now be used to calculate the design and operating parameters of an AHP to be implemented in a Kraft process. Supporting data for this study were taken from the work by Bakhtiari et al. [29], in which implementation of AHPs in a Kraft pulping process was studied using a new methodology for the optimal integration of those devices in a process [30]. It provides systematic guidelines for the proper selection of heat sources and sinks to maximize the benefit derived from heat pumping, while respecting both process constraints and operating requirements of the AHP. A simple case has been chosen to illustrate the capability of the model in the design mode.

Bakhtiari et al. [29] showed that in the Kraft process, there is a very large amount of energy at low temperatures which has the potential to be upgraded and recovered. In the selected example, the particular objective of the heat pump is to upgrade the low temperature heat load from the effluent of the pulp bleaching plant to a higher temperature level. Hot water from the power plant was selected as the proper above the pinch point. Low pressure steam (LP) at 345 kPa was used as the driving energy for the generator.

Table 6-6 presents the properties of the selected streams.

Table 6-6 Properties of the selected streams [29]

Stream	T_{in} (°C)	T_{out} (°C)	Flow (kg/s)	Heat load (kw)
Bleaching Alkaline effluents	56.1	44	27.8	961
Hot water	57	76	19	2229
LP steam	144	144	0.63	1268

The model calculates the design parameters of the heat pump in a steady state operation. It was assumed that the four heat exchangers are shell-and-tube, and ε_{SHX} and x_{st} are respectively 0.7 and 59 %, that safely is far from the crystallization zone. The bleaching alkaline effluents are cooled in the evaporator from 56.1 °C ($T_{e,in}$) to 44 °C ($T_{e,out}$), using 0.63 kg/s LP steam as the driving energy in the generator ($T_{g,in} = T_{g,out} = 144$ °C) . Condenser and absorber release the useful energy above the pinch point to increase the temperature of 19 kg/s of hot water from 57 °C ($T_{a,in}$) to 77 °C ($T_{c,out}$). The design and dimensioning parameters are presented in Table 6-7 as the input and calculated values. For each heat exchanger, the total surface area required is calculated and the number of tubes is obtained for the given values of the physical dimensions of each unit. Figure 6-9 presents the cycle in the phase diagram of the binary solution. It shows that the desired temperature lift can be accomplished in a single effect machine, with reasonable temperatures, pressures and concentrations. Mass flow rate and concentration of the weak solution are calculated at 3.9 kg/s and 53%; respectively. A typical value of 1.76 is obtained for the $COP_{heating}$..

Table 6-7 Design parameters for the single stage AHP

	Description	Unit	Absorber	Evaporator	Condenser	Generator
Input	Q	kW	1213	961	1016	1268
	Inside diameter of tube	mm	13.84	13.84	13.84	13.84
	Outside diameter of tube	mm	15.87	15.87	15.87	15.87
	Length of the unit	m	6.096	6.096	6.096	6.096
	Number of tubes in the vertical direction per pass		8	8	12	12
	Number of passes		4	4	1	2
	Fouling factor	m^2°C /Kw	9E-6	9E-6	9E-6	9E-6
Computed	Pressure	KPa	8.3	8.3	54.5	54.5
	U	kW/m^2°C	0.8	2.3	2.1	1
	LMTD	°C	19.5	5.6	11.4	15.3
	Calculated Surface area	m^2	77.7	74.6	42.4	82.9
	Total number of tubes		256	256	144	288
	Number of tubes in the horizontal direction per pass		8	8	12	12

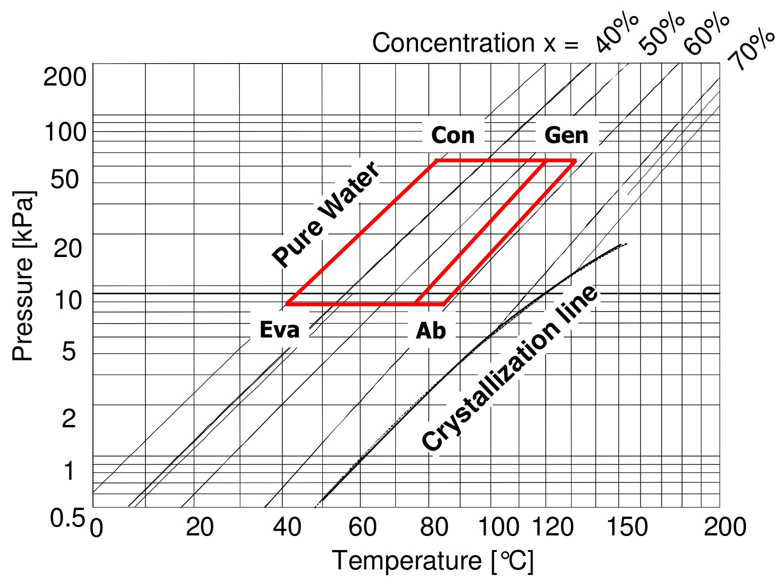

Figure 6-9 The proposed AHP in LiBr/H_2O equilibrium diagram

6.9 Conclusion

In this work, a single-stage H_2O-LiBr absorption chiller of 14 kW was experimentally characterized and modeled. The machine performance, as described by cooling capacity and COP were measured at different temperatures of chilled, cooling and hot water and, different flow rates of cooling and hot water.

The results have shown that an AHP has almost a constant COP over a large hot water inlet temperature which makes such a device well suited for trigeneration or solar cooling applications. The COP is primarily influenced by the cooling stream temperature and flow rate. The heat pump cooling capacity is more sensitive to cooling stream and generator inlet temperature than it is to chilled stream temperature.

A model for analysis and design of H_2O-LiBr absorption has also been developed. Good agreement of the experimental and simulation results was observed. The model has the capability to use either actual heat transfer coefficients computed experimentally or to estimate them by theoretical or empirical correlations. The model is capable of quick and reliable simulation and can be used as an aid to design of absorption cycles. It is shown how the developed model can be linked to the AHP implementation methodology for the design of an AHP to be implemented in an existing process. The design parameters are presented.

The development of economic data to complement the technical design capability of the model would constitute a very valuable tool for the optimal design of AHPs. It would assist engineering work for the implementation of such machines in industrial processes.

6.10 Acknowledgments

Max Siegel research intern and Keyvan Bararpour MSc student, contributed to the experimental work presented and to the developing the thermodynamic model; their contributions are acknowledged. This work was supported by research grants from the Natural Science and Engineering Research Council of Canada.

6.11 Appendices

6.11.1 Appendix A

Energy equations:

Condenser:

$$Q_c = m_c(h_{c,out} - h_{c,in}) = m_{ref}(h_7 - h_8) = U_c A_c LMTD_c \tag{A.1}$$

$$LMTD_c = \frac{(T_{c,in} - T_8) - (T_{c,out} - T_8)}{\ln \dfrac{T_{c,in} - T_8}{T_{c,out} - T_8}} \tag{A.2}$$

Evaporator:

$$Q_e = m_e (h_{e,in} - h_{e,out}) = m_{ref} (h_{10} - h_9) = U_e A_e LMTD_e \tag{A.3}$$

$$LMTD_e = \frac{(T_{e,in} - T_{10}) - (T_{e,out} - T_9)}{\ln \dfrac{T_{e,in} - T_{10}}{T_{e,out} - T_9}} \tag{A.4}$$

Generator:

$$Q_g = m_g (h_{g,in} - h_{g,out}) = m_{ref} h_7 + m_{st} h_4 - m_{we} h_3 = U_g A_g LMTD_g \tag{A.5}$$

$$LMTD_g = \frac{(T_{g,in} - T_4) - (T_{g,out} - T_7)}{\ln \dfrac{T_{g,in} - T_4}{T_{g,out} - T_7}} \tag{A.6}$$

Absorber:

$$Q_a = m_a (h_{a,out} - h_{a,in}) = m_{ref} h_{10} + m_{st} h_6 - m_{we} h_{10} = U_a A_a LMTD_a \tag{A.7}$$

$$LMTD_a = \frac{(T_6 - T_{a,out}) - (T_1 - T_{a,in})}{\ln \dfrac{T_6 - T_{a,out}}{T_1 - T_{a,in}}} \tag{A.8}$$

Solution heat exchanger:

$$Q_{SHX} = m_{we} (h_3 - h_2) = m_{st} (h_4 - h_5) = U_{SHX} A_{SHX} LMTD_{SHX} \tag{A.9}$$

Overall:

$$Q_g + Q_e = Q_a + Q_c \tag{A.11}$$

Mass balances:

$$m_{we} = m_{st} + m_{ref} \tag{A.12}$$

$$x_{we} m_{we} = x_{st} m_{st} \tag{A.13}$$

6.11.2 Appendix B

Uncertainty analysis:

Kline and McClintock method for estimating the uncertainty in experimental results is based on specifications of the uncertainties in the primary experimental measurements.

If F is a given function of m independent variables $y_1, y_2 \ldots y_m$,

$$F = F(y_1, y_2, \ldots, y_m) \tag{B.1}$$

And $un_1, un_2 \ldots un_m$ are the related uncertainties of the independent variables then the uncertainty in the result (un_F) is given as:

$$un_F = [(\frac{\partial F}{\partial y_1} un_1)^2 + (\frac{\partial F}{\partial y_2} un_2)^2 + \ldots + (\frac{\partial F}{\partial y_n} un_n)^2]^{0.5} \tag{B.2}$$

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GENERAL DISCUSSION

Implementation Guidelines

The first part of the thesis consisted of the development of a systematic methodology for the integration of AHPs in a process. The methodology of integration of traditional heat pumps such as vapour recompression heat pumps or electrically driven compression heat pumps is well known and discussed in the literature. However integration of more complex configurations like absorption heat pumps and absorption heat transformers in a process has not been thoroughly investigated. Guidelines are formulated for the proper selection of heat sources and sinks that will maximize the benefit derived from heat pumping while respecting process constraints and operating requirements of the AHP.

Based on Pinch Analysis, the composite curve diagrams are used to position heat pumps in the process so as to maximize the overall energy benefit. The composite curves diagram are preferred rather than the grand composite curve diagram, because it represents all streams involved in the process and thus, later can be used to redesign the HEN.

The method assumes that a Pinch Analysis of the process has been performed and thermal data on all streams are available. Since heat pumping is more expensive than heat exchanging, the HEN should first be considered and the process should be heat-integrated. It is recommended that other energy recovery methods such as condensate return and water closure be performed before pinch analysis and heat pump implementation. In general, they are cheaper energy recovery measures and they help reduce the pinch point temperature, which facilitates the implementation of AHPs. The developed methodology is summarized in the organigramme of Figure 7-1.

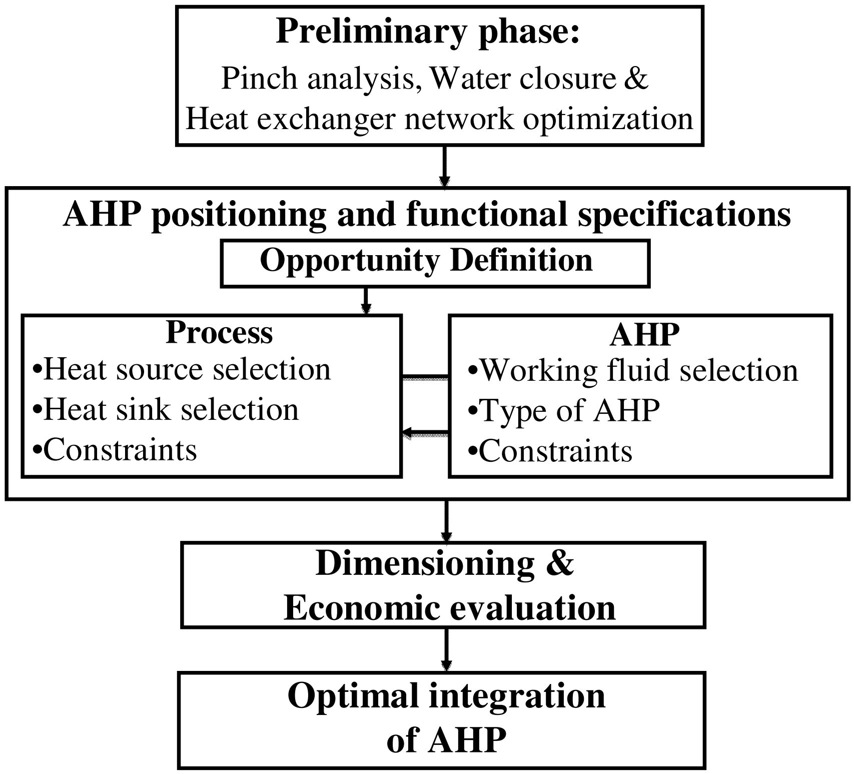

Figure 7-1 Methodology of integration of AHPs

The proposed method introduces several features that reveal superior designs to what could be achieved by either of the two conventional methods for designing AHPs and integration of traditional compression-based heat pumps. Indeed, it considers the interactions between the AHP and the process. It determines the streams that will best fit the AHP, and the best configuration and working pair, taking into account working fluid phase diagrams and other design and technical constrains.

A case study is presented and it was shown that even for a heat-integrated process, there can still be room for additional utility savings.

In this work only pinch-correct heat exchanger network reconfigurations have been considered. In further studies, the possibility to reduce the heat exchanger network investment cost by allowing balanced cross-pinch heat exchange should also be considered. This rearrangement might results in a better process integration in terms of overall process revenues as well as simpler and more efficient (higher COP) heat pump design.

A process working with non-steady-state conditions may change the operating conditions for heat pump, i.e. change in COP or amount of delivered heat. The developed methodology does not consider such conditions. In such cases, the heat exchanger network should be flexible enough to handle such derivations by allowing the heat load changes in the absorption heat pump to be shifted to the cold and hot utilities. Although, it was shown (chapter 6) that the absorption heat pump performance does not vary much with the temperature changes in the system and it could dampen both the temperature and heat load oscillations in the process.

Challenging the methodology in other industrial processes should be performed in order to identify limits of the methodology and enhance its robustness. For the processes with open composite curves (in which the composite curves are relatively far from each other as described by (Wallin et al., 1990)) and low pinch point temperature (which results in lower temperature lift and practical AHP design), absorption heat pumps can provide a definite potential to decrease the annual operating costs. With closer composite curves and high pinch point temperature, the possibilities of successfully installing an absorption heat pump are not that obvious, but the potential should be considered in every case. In such cases, other special working fluid pairs, such as sulphuric acid and various AHP configurations should be considered.

Based on the results presented in this work and scientific literature (Wallin *et al.*, 1990; Wallin and Berntsson, 1994), the most important factors for the implementation of absorption heat pumps are the composite curve, the electricity-to-fuel price ratio, the pinch point temperature, the absolute level of energy prices, the price of a heat exchanger and the specific absorption heat pump cost. A case which looks not practical and feasible in one place or time might be the best choice in another place or time. For example, with the future technology development, a single stage AHP with a higher COP will be preferred instead of implementing a double lift AHP in the first identified opportunity in the Kraft process.

Opportunities in the Kraft Process

There are only few reports of actual implementation of AHPs in the pulp and paper industry in the scientific literature and, none of these reported feasibility studies are based on the Pinch Analysis.

In this research, the potential use of AHPs for heat upgrading in the pulp and paper industry is demonstrated and validated. It shows that even for a fully energy and water optimized mill, there is still a potential for further utility savings. Two generic opportunities have been identified for the Kraft process that involve a broad spectrum of machine configurations, mode of operation and integration context.

In the studied Kraft process, there is a very large amount of energy at low temperatures which has the potential to be recovered. The bleaching plant is among those plants with the greatest potential for heat upgrading. Analysing the optimized heat exchanger network, it is observed that most of the available heat sinks above the pinch point are at relatively high temperature levels. To avoid the required large temperature lift and to have practical AHP design, the optimized heat exchanger network was reconfigured. The new configuration caused the available heat sinks to be at temperatures closer to the pinch point; a key feature of the most profitable retrofits. It resulted in a slight increase of the total heat exchanger area (capital cost), but the utility demand remained constant. This reconfiguration only marginally increased the cost of the heat exchangers required. Heating and cooling requirements will be further reduced by the installation of AHPs.

The first identified opportunity deals with the production of chilled water in the chlorine dioxide (ClO_2) making plant. In the current process configuration of the mill, water chilling is accomplished by a commercial compression heat pump and, by a steam ejector which is operated with MP steam. Considering the developed methodology, it was shown that the desired chilled water load and temperature can be achieved by means of a double lift machine using the $LiBr/H_2O$ working pair. In this case, a combination of heat sink streams above the pinch point is selected. The net hot and cold energy demand reductions represent 4 and 17 % of the MHR and the MCR, respectively.

The second opportunity is also identified in the pulp bleaching plant. Under the current process configuration, bleaching effluents, which is a heat source below the pinch point, is sent to the sewer. Single stage AHP using $LiBr\text{-}H_2O$ as the working pair, was the selected configuration in this case. The net hot and cold energy demand reductions represent 2 and 8.6% of the MHR and MCR, respectively.

An economic evaluation was performed to determine the economic feasibility of both AHP implementations. Simple pay back times of 2.7 and 1.7 years have been estimated for the two presented cases. Life-cycle cost analysis of the two cases was also performed using the net present value method. Figure 7-2 shows the evolution of the NPV over the investment period.

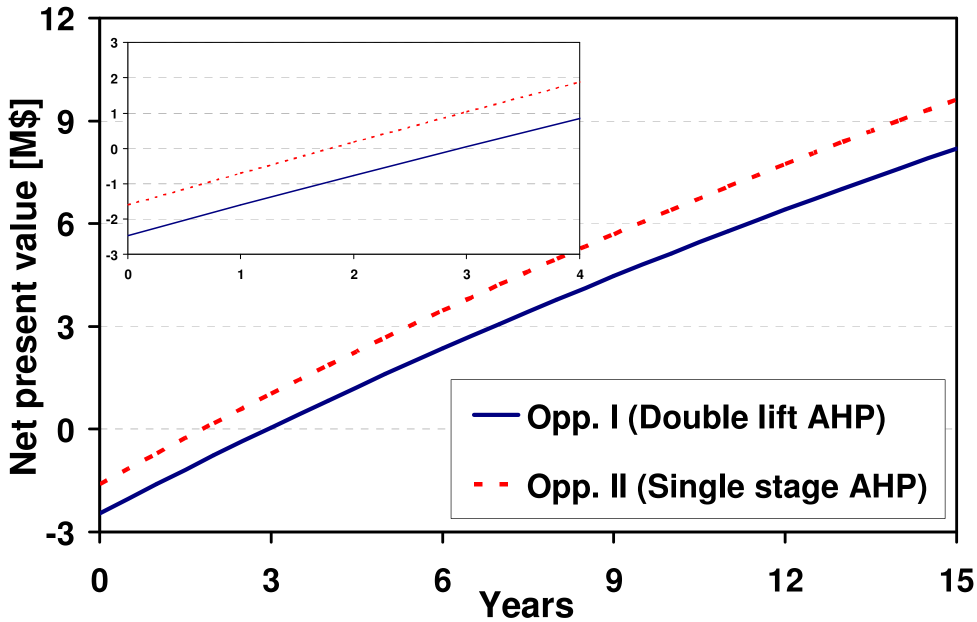

Figure 0-2 Net present value for the two general opportunities

Although the identified opportunities were presented for a particular Kraft mill, the idea of implementing an AHP in the bleaching plant to produce chilled water and the use of wasted energy in the effluents could be extended to all Kraft mills and therefore became generic implementations. Specific heat sinks above the pinch point and AHP designs might vary from one plant to another, but one should always be able to identify and optimize them based on the presented methodology.

Modeling and Experimental Evaluations

A single-stage H_2O-LiBr absorption chiller of 14 kW was experimentally characterized and modeled. The machine performance, as described by cooling capacity and COP were measured at different temperatures of chilled, cooling and hot water and, different flow rates of cooling and hot water.

A model for analysis and design of H_2O-LiBr absorption has also been developed. The model has two different fields of applications. It can be used either to analyze the performance of an existing machine under various operating conditions or to design and dimension a new one.

The results have shown that an AHP has almost a constant COP over a large hot water inlet temperature, which makes such a device well suited for trigeneration or solar cooling applications. The COP is primarily influenced by the cooling stream temperature and flow rate. The heat pump cooling capacity is more sensitive to the cooling stream and the generator inlet temperature than it is to the chilled stream temperature.

The model has the capability to use either actual heat transfer coefficients computed experimentally or to estimate them by theoretical or empirical correlations. It was shown that the COP is relatively insensitive to changes in U while the capacity is more sensitive to those coefficients.

It is shown how the developed model can be linked to the AHP implementation methodology for the design of an AHP to be implemented in an existing process. The design parameters are presented.

CONCLUSIONS AND RECOMMENDATIONS

The main objective of this thesis was to develop a methodology for the positioning and the dimensioning of absorption heat pumps (AHP) in processes. A general methodology was developed that is applicable to new or existing processes. The methodology respects the process constraints as well as AHP thermal and design constraints.

A model for analysis and design of H_2O-LiBr absorption has also been developed and it was shown how the developed model can be linked to the implementation methodology.

More specifically, the methodology allowed for the demonstration of a cost effective and a technically feasible retrofit of AHPs in an existing Kraft mill. Two generic opportunities have been identified that could be extended to all Kraft mills.

Original Contributions

A numbers of contributions have been brought to the field in the course of this work:

- The development of a novel implementation methodology for AHPs allowing for the retrofit of AHPs in any process. The method has the original novelty of including:
    - Pinch Analysis as the reference information
    - Process constraints
    - AHP thermal and design constraints
    - Interactions between the two (process and AHP)

- Development of an AHP design model.
- Application of the methodology to a Kraft process to determine generic opportunities, design the appropriate AHP and position them optimally in the process.

Recommendations for Future Research

Regarding future work, there are many possible directions. The developed methodology could be applied to other types of processes (e.g. chemical and petrochemical sectors, food processing sectors) in order to identify optimal opportunities in those sectors. Such studies should also include other types of heat pumps such as compression heat pumps and absorption heat pumps using other working fluids.

In this work only the two common working fluid pairs (NH_3-H_2O and LiBr-H_2O) are considered. Some alternative working fluids are also available for specialized application. The reason why none of these alternative fluids have gained a market is that the combination of properties exhibited by the conventional fluids is hard to compete against. However, for specialized application such as high temperature, other working fluid pairs should also be considered in the study.

Unified methodology for analysing combined cooling, heating and power should be developed. In that methodology, different types of heat pumps should be considered and the potential for heat pumping and increased heat exchange should be compared.

To develop an experimental and analytical study of the generator in order to characterize the heat and mass transfer coefficients in that components.

Another major area for future research would be the development of economic data to complement the technical design capability of the model which would constitute a very valuable tool for the optimal design of AHPs. It would assist engineering work for the implementation of such machines in industrial processes. So far, there is no reliable cost estimation model for AHPs, considering different configuration and working pairs and this work has shown that there is a need for the development of such a tool.

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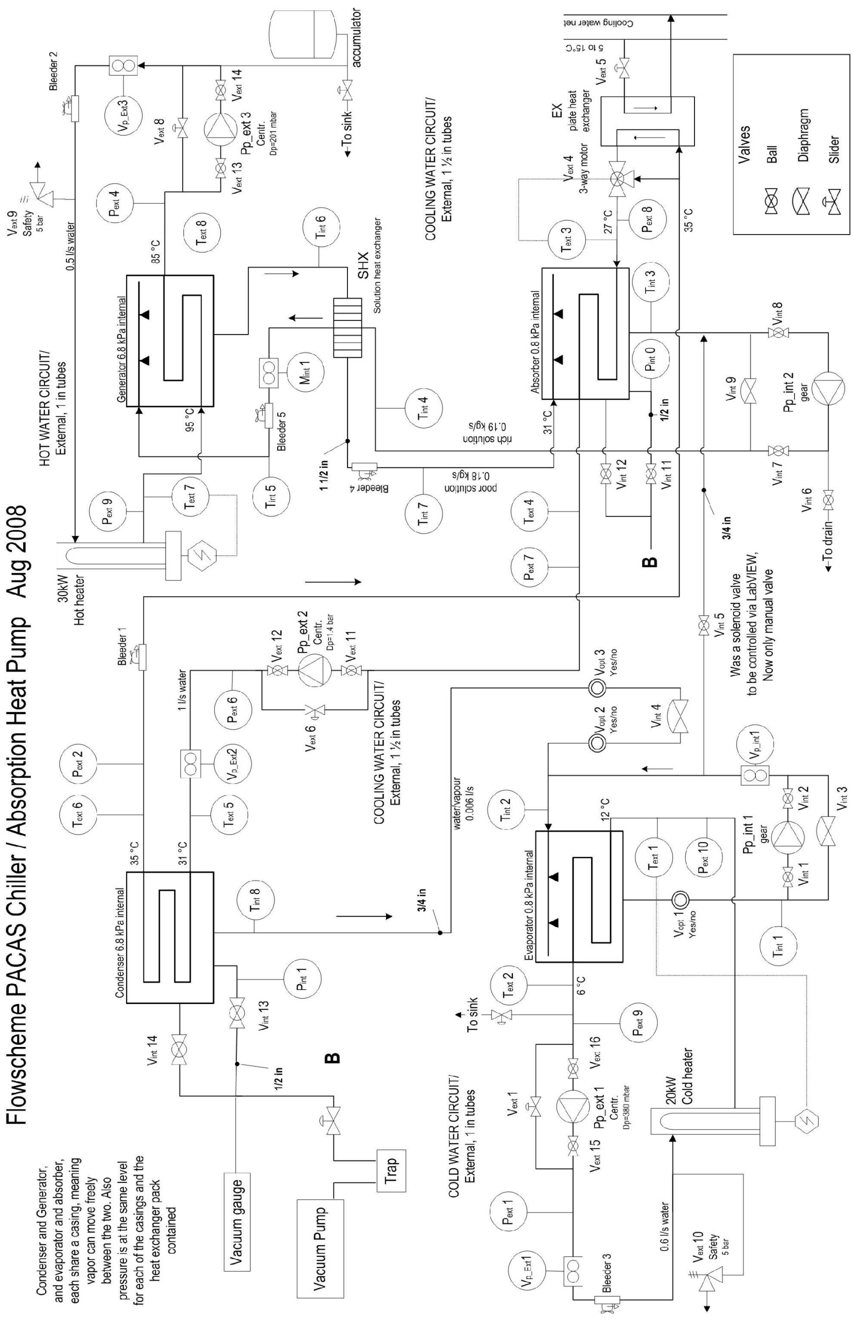
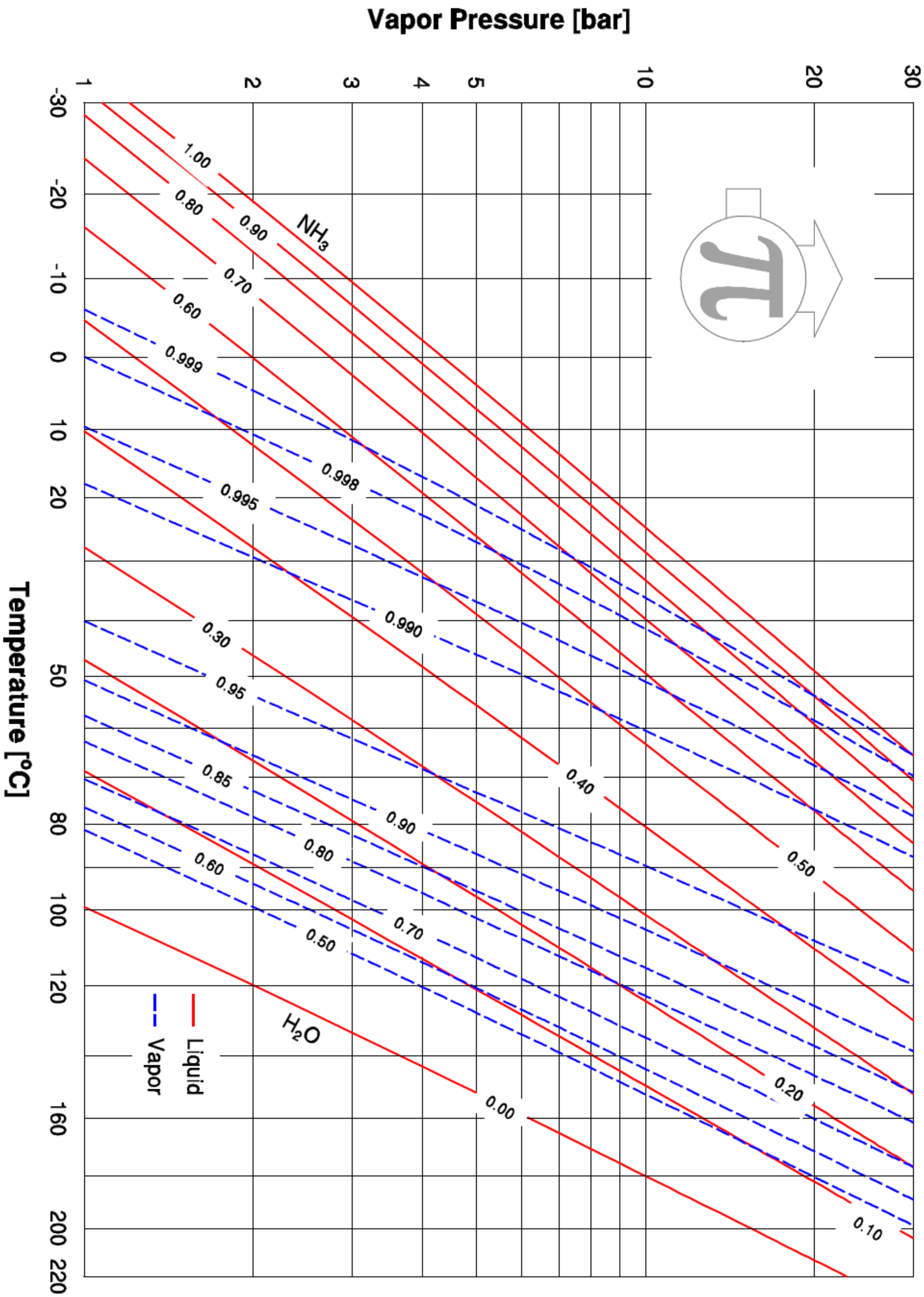
APPENDICES

A-1 Absorption heat pump flow sheet (prototype at Polytechnique)



A-2 Van t'Hoff diagram for solution of ammonia/water (Conde M. Engineering, 2006)



A-3 Van t'Hoff diagram for solution of LiBr/water